BACHELOR THESIS – ME141502

ANALYSIS OF ENERGY BALANCE FOR MAN 6L 23/30 FOUR STROKE DIESEL ENGINE

STEPHANUS CHRISTO P.
42 12 101 031

SUPERVISIOR:
Prof. Dr-Ing. Karsten Wehner
Dr-Ing. Wolfgang Busse

DOUBLE DEGREE PROGRAM
MARINE ENGINEERING DEPARTEMENT
FACULTY OF MARINE TECHNOLOGY
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ANALISA ENERGY BALANCE PADA MESIN DIESEL EMPAT TAK MAN 6L 23/30

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APPROVAL FORM

ANALYSIS OF ENERGY BALANCE FOR MAN 6L 23/30 FOUR STROKE DIESEL ENGINE.

BACHELOR THESIS

Submitted to Comply One of The Requirements to Obtain a
Bachelor Engineering Degree
in
Laboratory of Marine Power Plant (MPP)
Double Degree Program Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

Prepared by
Stephanus Christo P.
NRP: 4212 101 031

Approved by Supervisors of Bachelor Thesis:

1. Dr-Ing. Wolfgang Busse
2. Ir. Dwi Priyanta, M.SE

SURABAYA
JULY, 2016

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NRP. 4212 101 031

Approved by Head of Department of Marine Engineering

KEMENTERIAN RISET TEKNOLOGI DAN PENDIDIKAN TINGGI INSTITUT TEKNOLOGI SEPULUH NOPEMBER FAKULTAS TEKNOLOGI KELAUTAN JURUSAN TEKNIK SISTEM PERKAPALAN

Dr.Eng. Muhammad Badrus Zaman, S.T., M.T.
NIP. 1977 0802 2008 01 1007

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Prepared by
Stephanus Christo P.
NRP: 4212 101 031

Approved by representative of Hochschule Wismar in Indonesia:

Dr-Ing. Wolfgang Busse

HOCHSCHULE WISMAR
UNIVERSITY APPLIED SC TECHNOLOG BUSINESS AND DESI

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TASK DESCRIPTION

Thema: **Energy balance for four stroke diesel engine on example at MAN 6 L 23/30**

Student:

Supervising Professor:	Prof. Dr.-Ing. Karsten Wehner	Hochschule Wismar
Assistent Supervisor:	Dr-Ing. Wolfgang Busse	Hochschule Wismar

Date of issue:	April 2016
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The energy efficiency balance of internal combustion engines is one of the most important parameter to assess the quality of these engines. The input energy in fluid form contains chemical bounded energy, which will be transformed inside the engine in heat and then in mechanical energy. While the heat and the mechanical energy output can be measured exactly, the exhaust gas needs to consider separately. Because, exhaust gas is a carrier of chemical bounded energy like CO and SO as well as soot. On the other hand it contains a number of gas components, which need energy for the creation process like different sorts of nitrogen oxides. The object for this task is the laboratory main engine MAN 6L 23/30. Relevant data can be taken from existing measurements but are also to measure by new test bed drives.

The following aspects should be particularly considered:

1. Development of an complete energy balance processes
2. Verifying of the measuring method for fuel quantity
3. Development of an algorithm based on thermodynamically equations to determine the energy content in exhaust gas including the chemical bounded "lost energy"
4. Development of an algorithm to calculate the energy of endotherm reactive gases NOx

The supervising Professor reserves the rights to extend or to narrow down the scope of the task during processing. Establishing contacts with other institutions and companies must be agreed with the supervisors. The publication of the work or parts of it requires the prior permission of the supervisor. The work shall be prepared in accordance with the applicable guidelines of Hochschule Wismar for academic and scientific work. At least two consultations with the supervising Professor are required as part of the processing. The finished work is to be submitted in electronic form and in four printed copies in the organization office in Warnemünde.

Assistant Supervisor

Prof. Dr.-Ing. K. Wehner

Dr-Ing. Wolfgang Busse

HOCHSCHULE WISMAR UNIVERSITY OF APPLIED SCIENCES TECHNOLOGY, BUSINESS AND DESIGN

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NRP	:	4212101031
Bachelor Thesis Title	:	Analysis of Energy Balance for MAN 6L 23/30 Four Stroke Diesel Engine
Department	:	Marine Engineering

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Surabaya, July 2016

Stephanus Christo Probojati

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ABSTRACT

Name : **Stephanus Christo P.**
NRP : **4212101031**
Department : **Marine Engineering**
Supervisors : **Prof. Dr-Ing. Karsten Wehner**
Dr-Ing. Wolfgang Busse

Energy efficiency of thermal engines is a relationship between the total energy contained in the fuel and the amount of energy that converted to perform a useful work. It is concerned with the transformation of energy from fuels into conveniently used forms such as rotational energy, electrical energy, heating, and cooling. To gain a high efficiency there must be a higher energy converted into a useful work and the energy lost must be decreased. Therefore, the energy balance was made to analyze the distribution of energy inside the engine

The distribution of energy from fuel mainly flows to the exhaust gas, cooling system, lubricating oil system, and the most important to the engine crankshaft. Other than that, there are parts energy from fuel that can't be converted into useful work due to the formation of several substances by chemical reaction and incomplete combustion, which is called chemical losses. Incomplete combustion occurs when there is insufficient oxygen to allow a hydrocarbon fuel source to react completely with oxygen to produce carbon dioxide and water. And it would produce the unburned fuel material and CO gases. Incomplete combustion would decrease the thermal efficiency of the diesel engine. And chemical losses could be an indicator whether one engine is under good performance or not. Other than losses already mentioned above, there is also energy lost due to the formation of NOx. It means there is part of the energy from fuel that converted into heat and produces NOx. Thus, it is important to make an energy balance and analyze how big the energy which can be converted into useful work.

The project's approach involved laboratory diesel engine testing and analysis of combustion products inside exhaust gas. The distribution of energy from fuel was measured and calculated to determine the energy balance of diesel engine. And the primary output from this bachelor thesis was detailed energy audit across a variation of engine speed-load conditions.

Keywords: *Diesel Engine, Energy Balance, Thermal losses, Chemical Losses, Energy Audit*

ABSTRAK

Nama : Stephanus Christo P.
NRP : 4212101031
Jurusan : Marine Engineering
Pembimbing : Prof. Dr-Ing. Karsten Wehner
Dr-Ing. Wolfgang Busse

Energi effisiensi dari suatu mesin termal merupakan hubungan antara energi total yang terakndung di dalam bahan bakar berbanding dengan energi yang dikonversi menjadi useful work. Untuk menghasilkan efisiensi yang tinggi dibutuhkan lebih besar energi yang terkonversi menjadi useful work. Maka, energy balance dibutuhkan untuk menganalisa distribusi energi pada suatu mesin.

Distribusi energi dari bahan bakar, secara umum, terkonversi ke sistem gas buang, sistem pendingin, sistem pelumasan, dan yang paling utama adalah useful work. Selain itu, terdapat energi yang tidak dapat dikonversikan menjadi usefulwork yang disebabkan karena pembakaran yang tidak sempurna dan juga pembentukan beberapa partikel oleh reaksi kimia. Hal ini disebut chemical losses. Pembakaran yang tidak sempurna umumnya terjadi saat tidak terdapat oxygen yang cukup untuk melaksanakan proses oksidasi dari partikel karbon. Dan proses ini akan menghasilkan unburned material dan juga gas CO. Pembakaran yang tidak sempurna akan mengurangi efisiensi thermal dari suatu mesin. Dan chemical losses dan menjadi suatu indikator apakah suatu mesin berada dalam kondisi prima atau tidak.

Laporan ini melibatkan engine test di dalam laboratorium dan analisa produk pembakaran di dalam gas buang. Distribusi energi dari bahan bakar akan diukur dan dianalisa pada beberapa daya mesin untuk menentukan energy balance yang paling optimal. Dan output utama dari laporan ini adalah detil audit energi pada beberapa daya dan putaran mesin.

Kata Kunci: *Mesin Diesel, Energy Balance, Audit Energi, Chemical losses, Efsiensi Mesin***.**

PREFACE

When I was a freshman in university, there was a proverb that said:

"*Theory is when you know everything but nothing works.*
Practice is everything works but no one knows why.
In our laboratory, theory and practice are combined and the results is: Nothing works and nobody knows why"

And Luckily for me, I ended up to work in a laboratory to develop my bachelor thesis. As a student, I spent more than 80% of my study period by learning in a classroom and not had a really much experience to facing a real system. When I developed my bachelor thesis, I have to collect all the required data directly from the engine and facing a real system. And long story short, sometimes, I felt this proverb is actually worked. I have experienced that the results of calculation were not matched with what the theory have said. And sometimes they walk in opposite direction. And it took a time to find the reason why.

That is the impression that I got when I was working on my bachelor thesis in a laboratory. But, this experience actually made me become aware and curious about what the reason behind one theory. It made become aware of what are the factors in a real condition that could affect a formula which written in a textbook. To summarize, I have to say that I feel blessed to have experience working in a laboratory.

And also I would like to express my sincere gratitude to my supervisor Prof. Dr-Ing. Karsten Wehner and Dr.-Ing. Wolfgang Busse, for gave me a lot of guidance, knowledge, advice, and priceless opportunity. To Mr. Hartmut Schmidt and

Mr. Benjamin Muller who introduced me and guide me when I was working in a laboratory. Further, I would also like to acknowledge Ir. Dwi Priyanta MSE. Ph.D. as my lecturer who always give a lot of guidance and advice from the beginning of a semester. I realized that I would not have been possible to accomplish my bachelor thesis without all of them. Thank you.

Surabaya, July 2016

Stephanus Christo P.

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CHAPTER I
INTRODUCTION

1. 1 Background

The diesel engine was operated to achieve the desired power output, which is, for marine engineer’s purposes, to drive a ship at satisfied speed, and/or to provide electricity at a prescribed power. To determine this power he must allow not only for the cycle losses but for the friction losses in the cylinders, bearings, gearings, and heat lost by radiation. He must ensure that his engine could produce a prescribed power for operation. Thus, the energy balance of the diesel engine is required. The energy balance describes the distribution of energy from fuel that converts into useful work and several energy losses which exist in the form of exhaust gas, heat released to engine’s cooling system, heat released by radiation, friction losses, and also chemical losses due to incomplete combustion.

A typical diagram that shows the energy balance of diesel engine known as a Sankey Diagram. This diagram representing the various energy flows through a modern diesel engine. Thermal efficiency also can be determined from this diagram which equals to heat converted into useful work divided by total heat supplied. And thermal efficiency could be one of the indicators to determine whether one diesel engine operates under a good condition or not.

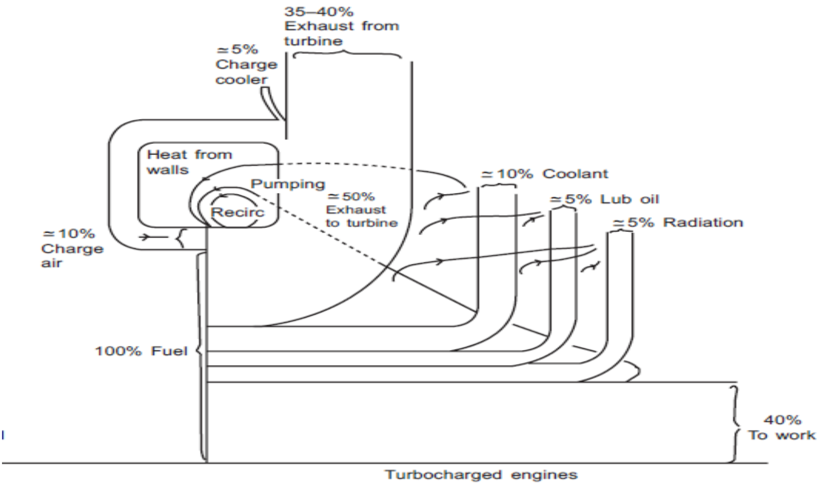


figure I.1 Sankey Diagram (Woodyard, Marine Dieesel Engines and Gas Turbines, 2004)

1.2 Problem Statements

A Marine diesel engine uses fossil fuels as the main source to produce energy that converted into useful work. This energy from fuel is converted into useful work that used to drive propeller of a ship. But, not all of the energy from a fuel can be converted into useful work. Fluid friction, mixing, and rapid expansion affects the amount of work extraction by the piston. The energy also lost in the form of exhaust gas, heat released to the cooling system, heat released by radiation, pumping losses, and also losses due to the friction of engine components. The other losses also occurred due to the incomplete chemical reaction of fuel which could be determined by analyzing the exhaust gas composition.

All of those losses would affect the brake thermal efficiency of marine diesel engine. Brake thermal efficiency is the overall measure of performance. In terms it is equal to heat converted into useful work divided by total heat supplied. Every engineer would maintain their engine by optimizing the injection settings, the air flow, coolant temperatures, at those values which give the best efficiency.

To determine brake thermal efficiency of the engine it is necessary to measure all of the energy input and energy output from diesel engine. Energy input could be measured from the amount of the fuel energy that used for combustion. While, the energy output was determined by measuring the useful work output and several losses from practical cycle of a diesel engine. Thermal efficiency could be one of the indicators to determine the performance of specific diesel engine. Thus, it is very important to measure the energy balance of diesel engine.

Based on the description above, the problem statement that can be concluded are:

1. How to determine the energy efficiency balance of engine MAN 6L 23/30?
2. How big the energy content in exhaust gas and what is the effect of chemical bounded energy losses to the overall energy efficiency?
3. In which engine load or operating point the energy distribution would be optimal?

1.3 Boundary Issue

The limitation of the problem of this bachelor thesis are:

1. The energy distribution of diesel engine would be measured and determine by experiment on the laboratory
2. The thermal losses would be measured and determined from the heat released to exhaust gas, cooling system,

lubricating oil system and input energy transferred to the fuel and inlet air

3. The chemical losses would be determined by analyzing unburned fuel and material inside the exhaust gas

1.4 Purposes of Bachelor Thesis

This bachelor thesis aims to:

1. Develop the complete energy balance of MAN 6L 23/30 four stroke diesel engine
2. Measure and analyze the thermal and chemical bounded energy losses of MAN 6L 23/30 four stroke diesel engine.
3. Analyze and make a recommendation in which engine load the energy balance would be optimal.

1.5 Benefits of Bachelor Thesis

The benefits could be obtained from this bachelor thesis are:

1. Understanding the concept of energy balance of internal combustion engines
2. Understanding energy distributions that occured inside internal combustion engines
3. Understanding the methods to develop the complete energy balance of internal combustion engines

CHAPTER II
STUDY LITERATURE

2.1 Energy Balance of Internal Combustion Engine

An Internal Combustion (IC) engine is a complex machinery, instrumentation, and services, all of which must work together as one system. Internal Combustion Engines, here, the hot combustion gases themselves become the working substance and energy is transferred directly from them as work. They can be reciprocating types, either it is the spark ignition engine or compression ignition engine (Milton E., 1995)

An IC engines can be considered as a thermodynamic ‘open system’, which is a powerful concept to understand the thermodynamic behavior of a system. It is linked to the idea of ‘control volume’, a space enclosing the system and surrounded by an imaginary surface often known as ‘control surface’. The advantage of this concept is that once has identified all the mass and energy flows into and out of a system, it is easy to visualize the inside picture of that system by drawing up an energy balance sheet of inflows and outflows. (Abedin, Masjuki, & Kalam, 2013)

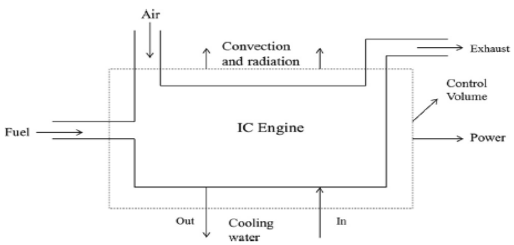


figure II.1 Control Volume of Internal Combustion Engine (Abedin, Masjuki, and Kolam, 2013)

2.2 Energy Balance Theory

Energy balance representing the flows of various energy through a diesel engine. Energy input was analyzed based on the mass of fuel injected into thecombustion chamber. While energies output were analyzed based on the produced useful work and other losses flows as heat and friction. The literature also shows that energy balance studies were done to optimize engine settings or system settings and it will determine in what range the engine will operate most efficiently (Kopac, Lutfi, & Mehmet, 2005).

The energy balance that used in this study review all followed the same basic principle. "Energy balance for a direct injection diesel engine shows that about one-third of fuel energy input is lost to the environment through heat transfer, another third is wasted as exhaust heat and only one-third is available as shaft work" (Sharma & Jindal, 1989). Thus, heat losses must be decreased to improve the engine efficiency and it is very important to know the fraction heat loss mechanism. The equation to calculates the energy balance was referred to the literature with a title of "Energy Balance on Internal Combustion Engines" (Van Gerpen, 1997), energy balance principle calculation that obtained from the literature of Hochschule Wismar and also combined with the other formula from other references that will be described in detail later.

1. Density of Humid Air

Firstly the ambient density of humid air must be determined based on the ambient pressure, temperature, and relative humidity. And the formula to calculates the density of humid air was given by the following equation (Weast, 1986, *Handbook of Chemistry and Physics*):

$$\boldsymbol{\rho\ humid\ air\ (\frac{kg}{m3}) = 1,229\ x \frac{273,15}{T\ amb}\ x \frac{P\ amb - 0,3783\ Pv}{1,013\ x\ 10^5}}$$

Equation 2.1 Density of Humid Air

Where P amb and T amb are the ambient pressure (Pa) and temperature (K) respectively. And Pv is the partial vapour pressurewhich depends on the relative humidity.

Where P amb and T amb are the ambient pressure (Pa) and temperature (K) respectively. And Pv is the partial vapour pressurewhich depends on the relative humidity.

2. Fuel Input Energy

The input energy going into the engine,*Qin,* was given by the equation:

$$\boldsymbol{Qin\ (\frac{kJ}{s}) = \dot{m}f\ x\ LHV}$$

$$\boldsymbol{Qin\ (\frac{kJ}{s}) = vf\ x\ \rho f\ x\ LHV}$$

Equation 2.2 Fuel Input Energy

Here, ṁf is the fuel mass flow rate and the LHV is the lower heating value or energy content of the supplied fuel.

The quantity of Lower Heating Value (LHV) is determined by subtracting the heat of evaporation of water vapor from higher heating value. LHV calculations assume that the water component is in a vapor state at the end of combustion, as opposed to the Higher Heating Value (HHV). The formula to determine the LHV is based on the EN ISO 3675, 1998 and given by the equation:

$$LHV\ (\frac{kJ}{kg}) = 4,19\ x\ (11040 - (2180\ x\ \rho\ fuel\ at\ 15°C) + 880\ x\ \rho\ fuel\ at\ 15°C)$$

Equation 2.3 Lower Heating Value

3. Brake Power Output

The brake power output, Pb, would be calculated with the following equation:

$$Pb\ (\frac{kJ}{s}) = \tau b\ x\ \omega$$

Equation 2.4 Brake Power Output

Here, τb, is the brake torque of the engine and ω is the angular velocity of the engine crankshaft.

4. Heat Released to Jacket Water

The energy transferred through engine coolant components is divided into three components which are jacket water cooler, lubricating oil cooler, and scavenge air cooler. To ensure the exact calculation of heat losses to the engine's cooling system, the calculation would be obtained on two sides of a heat exchanger which are the coolant side (fresh water side) and the fluids side (jacket water, scavenge air, and lubricating oil).

The heat losses to the jacket water system, Qc, are calculated by:

$$Qc\ (\frac{kJ}{s}) = \dot{m}c\ x\ Cpc\ x\ (Tc\ out - Tc\ in)$$

$$Qc\ (\frac{kJ}{s}) = vc\ x\rho c\ x\ Cpc\ x\ (Tc\ out - Tc\ in)$$

Equation 2.5 Heat Released to Jacket Water

Here vc and ρc are the volumetric rates and density jacket water respecitively at the average temperature. Cpc is the specific heat of the jacket water at temperature half way between jacket water inletand outlet temperature at the engine (Tc in and Tc out).The calculation also conducted on the freshwater side (coolant of jacket water cooler) and similar formula is used to determine the heat released bythe system.

5. Heat Released to Lubricating Oil

The amount of heat carried away by the lubricaing oil could be measured by the following equation:

$$\boldsymbol{Q\ oil}\ (\frac{\boldsymbol{kJ}}{\boldsymbol{s}}) = \dot{m}\ \boldsymbol{Coolant\ x\ Cp\ Coolant\ x}\ (\boldsymbol{T\ Coolant\ out} - \boldsymbol{T\ Coolant\ in})$$

$$\boldsymbol{Q\ oil}\ (\frac{\boldsymbol{kJ}}{\boldsymbol{s}}) = \boldsymbol{v\ Coolant\ x\ \rho\ Coolant\ x\ \ Cp\ coolant\ x}\ (\boldsymbol{T\ Coolant\ out} - \boldsymbol{T\ Coolant\ in})$$

Equation 2.6 Heat Released to Lubricating Oil

Here v and ρ are the volumetric flow rates and the density of the coolant at lubricating oil cooler respectively. Cp is the specific heat of the coolant at temperature half way between the temperature of the coolant coming out and going into the lubricating oil cooler(T coolant in and T coolant out).

6. Heat Released to Scavenge Air

Similiary, the input energy transferred to the scavenge air is given by:

$$\boldsymbol{Qa}\ (\frac{\boldsymbol{kJ}}{\boldsymbol{s}}) = \dot{m}\boldsymbol{a\ x\ Cpa\ x}\ (\boldsymbol{T\ air\ after\ cooler} - \boldsymbol{T\ air\ before\ cooler})$$

$$\boldsymbol{Qa}\ (\frac{\boldsymbol{kJ}}{\boldsymbol{s}}) = \boldsymbol{va\ x\ \rho\ air\ x\ Cpa\ x}\ (\boldsymbol{T\ air\ after\ cooler} - \boldsymbol{T\ air\ before\ cooler})$$

Equation 2.7 Heat Released to Scavenge Air

Here v and ρ are the volumetric flow rates and the density of the air at the ambient temperature. Cp is the specific heat of the air at temperature half way between inlet and outlet at the scavenge air cooler (T air before cooler and T air after cooler). The calculation also conducted on the freshwater side (coolant of scavenge air cooler) and similar formula is used to determine the heat released by the system.

7. Heat Released to Ambient Air

When an engine is being operated, there were some amount of heat that released to the ambient air by convection and radiation process. Thus, it becomes important to determine how much the energy released in order to produce an exact energy balance. The used method was measuring temperature between the surface temperature of engine block and ambient temperature. The detailed formula is shown below:

$$\boldsymbol{Q\ conv\ \left(\frac{kJ}{s}\right) = \frac{Tsurface - T\ amb}{Surface\ Area\ x\ h\ total}}$$

Equation 2.8 Heat Released to Ambient Air

Where h is the total convection and radiation heat transfer coefficient from cast iron (engine block material) to ambient air.

8. Thermal Losses of Exhaust Gas

The energy losses to exhaust gas exists in two forms, which are thermal and chemical losses. The thermal losses to the exhaust gas would be calculated with the following equation:

$$\boldsymbol{Q\ Exh\ Gas\ (\frac{kJ}{s}) = (\dot{m}\ f + \dot{m}\ Air) x\ Cp\ Exh\ x\ T\ Exh - \dot{m} Air\ x\ Cp\ Air\ x\ T\ Air}$$

Equation 2.9 Thermal Losses of Exhaust Gas

Here, ṁ f and ṁ arepresenting the mass flow rate of fuel and inlet air respectively. Cp Exh is the specific heat of exhaust gas at the mean temperature between exhaust temperature and ambient temperature.Specific heat of exhaust gas is determined with the following equation(Hochschule Wismar, Principle of Energy Balance Calculation)*:*

$$\boldsymbol{Cp\ Exh} = \frac{\dot{m}\boldsymbol{Exh\ min\ x\ Cp\ Exh}\,|_{T} + \dot{m}\boldsymbol{Air\ min\ x\ (\lambda-1)x\ Cp\ air}\,|_{T} + \boldsymbol{\lambda\ x\ XL\ x}\ \dot{m}\boldsymbol{Air\ min\ x\ Cp\ H2O}\,|_{T}}{\boldsymbol{1+\lambda\ (1+XL)}\dot{m}\boldsymbol{Air\ min}}$$

Equation 2.10 Specific Heat Of Exhaust Gas

All the specific heat (CP) that used in the above equation is obtained from table ... that shown in appendix. Lambda (λ) is the total air ratio that calculated with the following equation*(Principles of Energy Balance Calculation*, Hochschule Wismar*)*:

$$\boldsymbol{\lambda} = \frac{\dot{m}\ \boldsymbol{Air}}{\dot{m}\ \boldsymbol{Air\ minimum\ x}\ \dot{m}\ \boldsymbol{Fuel}}$$

Equation 2.11 Air Equivalence Ratio

Where ṁ Air minimum equal to the approximation of 14,53 of air / kg fuel. Andṁ Exh gas minumum equal to ṁAir minumum plus by 1. While XL is the water vapour content which depends on the ambient temperature and relative humidity. XL is calculated with following formula*(Principles of Energy Balance Calculation*, Hochschule Wismar*)*:

$$\boldsymbol{XL\ =\ 0,622\ \frac{Ps\ x\ \varphi}{P-\ Ps\ x\ \varphi}}$$

Equation 2.12 Water Vapour Content

Where Ps and $\boldsymbol{\varphi}$are the saturated steam pressure and humidity factor which obtained from table ... that shown in appendix.

9. Chemical Losses of Exhaust Gas

The chemical losses occurred due to the incomplete combustion of a diesel engine. And the chemical losses inside the exhaust gas would be determined by measuring the unburned fuel material, like HC, CO, C,Soot, and NOx gases inside the exhaust gas. The exhaust gas products analyzer will measure the unburned fuel material and NOx gases in the percentage of mass. To calculate the how big the chemical loss is, the percentage of unburned fuel material would be multiplied with its enthalpy at the ambient temperature. And the example of calculation is shown in the following formula:

1. $$\boldsymbol{HC = \ldots ppm \left(\frac{mg\ of\ HC}{kg\ of\ EG}\right)}$$
2. $$\boldsymbol{HC\ mass\ flow\ rates\ (\frac{gr}{s}) = HC\ ppm \left(\frac{mg\ of\ HC}{kg\ of\ EG}\right) x\ \dot{m}\ Air\ (\frac{kg}{s})}$$
3. $$\boldsymbol{energy\ losses \left(\frac{kJ}{s}\right) = HC\ mass\ flow\ rates \left(\frac{gr}{s}\right) x\ Enthalpy \left(\frac{kJ}{gr}\right)}$$

Equation 2.13 Chemical Losses of Exhaust Gas

While the chemical losses due to the CO gases would be calculated with the following formula*(Principles of Energy Balance Calculation*, Hochschule Wismar*)*:

$$\boldsymbol{Q\ CO\ (\frac{kJ}{s}) = 22{,}57 \left(\frac{131300}{LHV} - 1\right) x \left(\frac{CO}{CO2 + CO}\right) x \frac{Qin}{100}}$$

Equation 2.14 Chemical Losses due to CO gases

Where CO2 and CO is the percentaege of carbon dioxide and carbon monoxide gases inside the exhaust gas.

2.3 Exhaust Gas Emissions of Diesel Engine

Marine diesel engine emits an exhaust gas which contents particles such as nitrogen, oxygen, water vapor, carbon dioxide and also pollutant particles which consist of nitrogen oxides (NOx), sulphur oxides (SOx), and particulate matters (PM) (woodyard 2004). Those particles were produced due to the combustion of fuel in combustion chamber. Development of those particles required energy from the fuel and considered to be one of the factors that affect an energy balance of diesel engine.

The combustion quality of diesel engine could be analyzed by measuring the composition of their exhaust gas. The analysis of exhaust gas is important to determine whether the combustion occurs inefficient way or not. If it is found a lot of unburned fuel materials, like hydrocarbon (HC) and particulate matters, inside the exhaust gas it could be concluded that the combustion occurs in a poor condition. There are many factors that can lead to the poor combustion, for example, miss injection timing, defects of injection valves, bad fuel quality, etc.

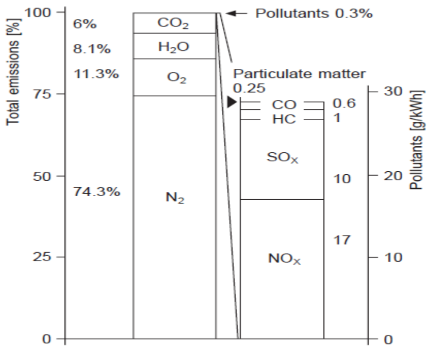


figure II.2 Typical Composition of the Exhaust Gas Products of a Medium Speed Diesel Engine (MAN B&W Diesel)

The NOx emission from diesel become a trending topic in maritime industries recently. It was due to a more strict regulation of NOx emission. Nitrogen oxides (NOx) are generated thermally from nitrogen and oxygen at high combustion temperature in the cylinder. There was some parts energy from fuel that converted into heat and produces NOx. And that part of the energy will affect the complete energy balance of diesel engine. Thus, it is become important to measure the energy required for the formation of NOx in order to develop an exact and complete energy balance of diesel engine.

2.4 Measurement Techniques

The energy distribution of diesel engine needs to be measured exactly in order to make a good energy balance. Thus, the good measurement techniques are required to produce the exact energy balance and later we could determine the quality of one specific engine. The measurement techniques that used in this bachelor thesis were similar with most of the energy balance studies that had been examined. And the energy distribution that needs to be measured are:

- **Density of humid air**
  The density of humid air would be determined by formula which depends on the ambient temperature, pressure and relative humidity.

- **Lower Heating Value**
  The LHV was determined by the methods from ISO EN 3675. This methodhas a formula that required a fuel density at 15°C. Firstly, the measurement was conducted by measuring the density of fuel sample at reference temperature and also determine the thermal

expansion coefficient (α) then these values would be used to determine the fuel density at 15°C.

- **Input energy from fuel**
  The input energy from fuel would be determined by measuring the fuel density and the volumetric flow rates of the fuel. And after that, it would be multiplied with the Lower Heating Value (LHV) which depends on the fuel composition. The LHV was determined by the methods from ISO EN 3675.

- **Break Power Output**
  Brake power output would be determined by measuring the engine's torque by means of dynamometers. Hydraulic dynamometers were used in most cases, however, it should be noted that the other dynamometers such as water brake, prony brake, eddy current, direct current, generator dynamometers could have been used (Winther 1975).

- **EnergyLosses as heat to cooling system (jacket water, lubricating oil, and scavenge air cooling system)**
  Energy Losses as heat to the three types cooling system, which are jacket water, lubricating oil and scavenge air cooling system, would be determined by measuring the volumetric flow rates of the fluids and the temperature differences between outlet and inlet at the engine's cooler. The fluids density at mean temperature between inlet temperature and outlet temperature also be used to converts volumetric flow rates into mass flow rates.

- **Thermal Energy Losses of exhaust gas**
  Energy lost to exhaust gas exist in two forms which are thermal and chemical losses. The thermal losses would be determined by determining the mass flow rates of the exhaust gas and multiplied it with specific heat and the temperature differences between exhaust temperature and ambient temperature. The specific heat of exhaust gas is determined at the mean temperature between exhaust and ambient temperature.

- **Chemical Energy Losses of exhaust gas**
  Chemical energy losses of exhaust gas were determined by measuring the unburned fuel material such as CO, hydrocarbon, soot and also the NOx gases which consist of NO2 and NO gases. The measurement was conducted by means of exhaust gas analyzer that could measure the composition of those substances in ppm (part per million) or percentage

CHAPTER III
METHODOLOGY

The methodology that used in this bachelor thesis are based on a literature studies and a diesel engine test inside machinery laboratory.

3.1 Problem Statement

The progress of bachelor thesis started with identifying and formulating the problems about the work to be carried and also the limiation or constraints of the issue. This step is aimed to simplify the issue.

3.2 Data Collection

The required data were the principle dimension of the engine and engine's parameters that would use as early data to develop a calculation program in order to determine the energy balance of diesel engine. All of the data was obtained from the marine diesel engine plant at Hochschule Wismar laboratory.

3.3 Development of Calculation Program

The calculation program was built based on the formula regarding energy balance of diesel engine, and the early data that used for the calculation is obtained from the previous engine test in the laboratory of Hochschule Wismar. And the calculation program was built by utilizing Microsoft Excel.

3.4 Requirement check I

Requirement check was aimed to check whether the calculation program has worked properly or not. The process would examine the formula that used in the calculation. If the formula is not written correctly, the process will be returned to the development of calculation program.

3.5 Laboratory Test Preparation

Laboratory test preparation includes planning and preparation of engine testing apparatus.

3.6 Fuel Oil Analysis and Lower Heating Value (LHV) Calculation

Fuel oil analysis was meant to determine the composition of fuel being used in order to calculate the Lower Heating Value of the fuel.

3.7 Measurement of Thermal and Chemical Losses of Diesel Engine

Thermal losses would be calculated by measuring energy lost in the exhaust gas, cooling system, lubricating system, radiation, and charge air. While, chemical losses would be determined by measuring the unburned fuel materials inside exhaust gas.

3.8 Energy Balance Analysis

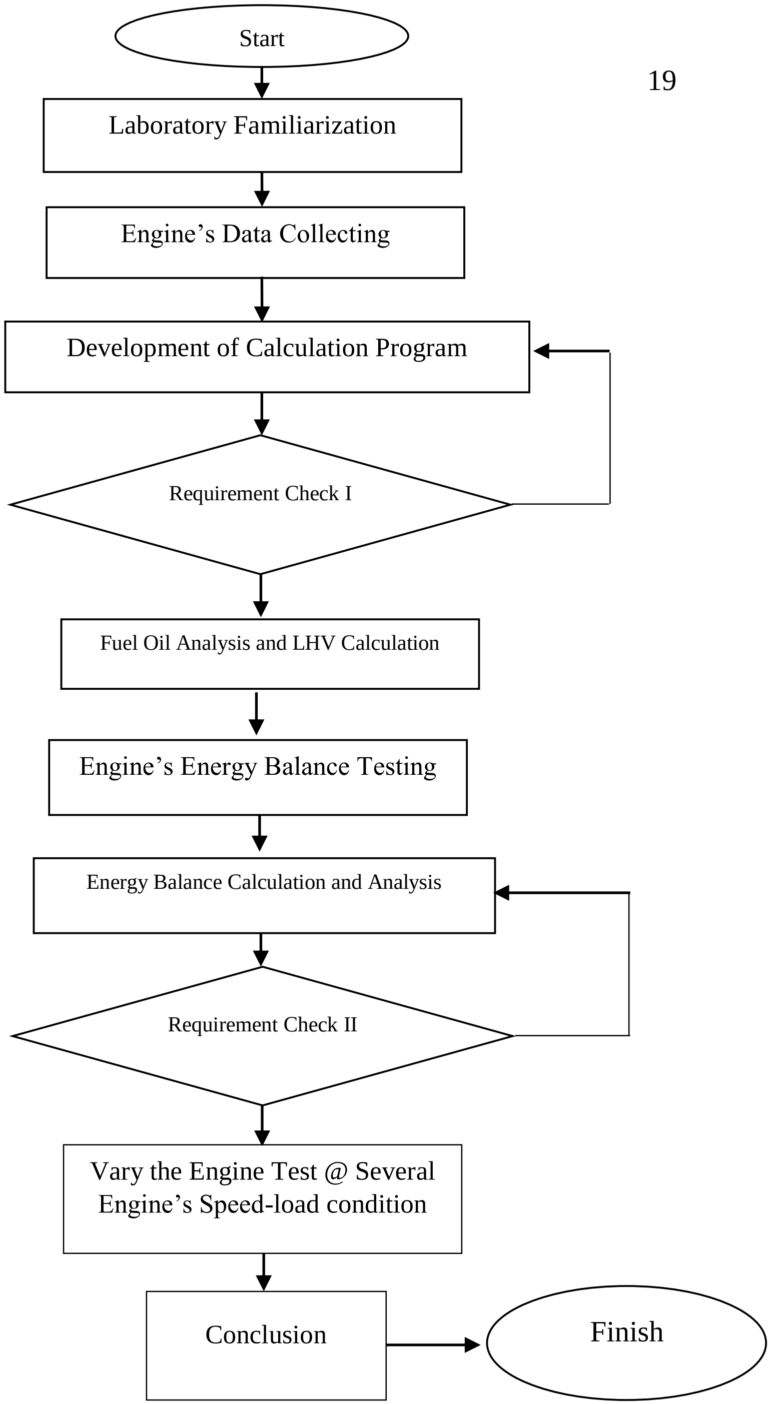
Energy Balance analysis would be conducted by utilizing the data from an experiment of thermal and chemical losses. The energy balance analysis would determine the total energy input and output through the diesel engine.

3.9 Requirement Check II

The energy balance would be checked whether the amount of energy output has a value equal to or near to the energy input. If the there were an excess gap, the data will be collected again by measurement of thermal and chemical losses.

3.10 Conclusion

The conclusion would be determined from the energy balance analysis. The final results are a diagram that representing the energy input and energy output of diesel engine and thermal efficiency across engine load condition.



Graph III.1Research Methodology

3.11 Engine Testing Apparatus

3.11.1 Test Engine Specification

In order to make an energy audit, a MAN 6L 23/30A-DKV was tested on an engine dynamometer test bench. It is a four stroke, 6 linear cylinders, turbocharged, 960 kW diesel engine. An energy audit would be determined by measuring all the energy distributions from the fuel to a certain power output of the engine. The engine specifications are shown in table 1 and the data are valid at the maximum continuous rating (MCR) of the engine with ambient air temperature 45°C and air pressure 1000 mbar.

Table III.1 Test engine specification

MANUFACTURER	MAN B&W
TYPE	6 L23/30A-DKV
BORE (mm)	225
STROKE (mm)	300
DISPLACEMENT / cyl (L)	11,9
RATED POWER (kW)	960
RATED SPEED (RPM)	900
MEAN EFFECTIVE PRESSURE (bar)	17,9
MAX COMBUSTION PRESSURE (bar)	135
COMPRESSION RATIO	12,5

figure III.1 Engine MAN 6L 23/30A-DKV

3.11.2 Control Volume of The Engine

The test engine was aimed to audit the energy flow in MAN 6 L23/30A-DKV diesel engines. In order to characterize the energy distribution, a suitable control volume is needed. Figure 3.2 shows the control volume which analyze all the energy distribution from the outlet of exhaust gas turbine until the inlet of the compressor on the air intake side of the engine.

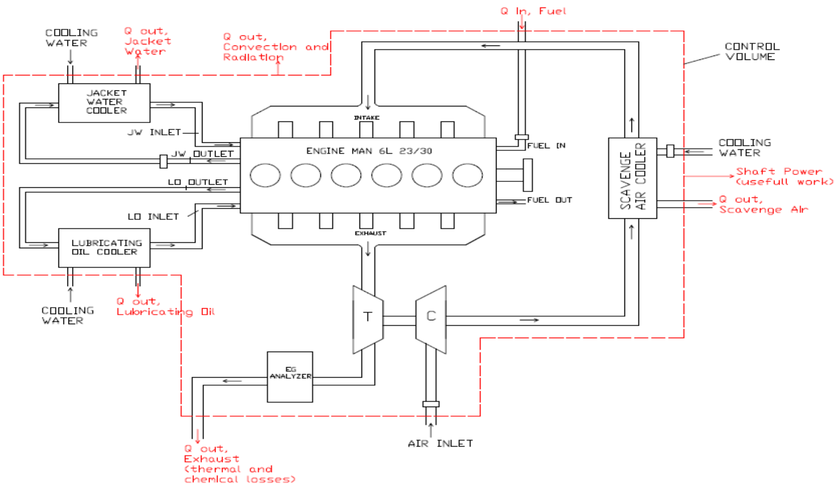


figure III.2 Control Volume of The Engine

3.11. 3 Layout of Engine Test

The engine instrumentation and testing procedure would be explained in this section. In order to determine the energy balance, the temperature in the air, exhaust, coolant, and oil streams were measured by the thermocouples which installed on all fluid flow pathways. And also all the volumetric flow rates of the fluid were measured by flowmeter that installed on its pathways. The exhaust gas analyzer was used to measure the particles inside the exhaust gas, in order to calculate the chemical energy lost inside the exhaust gas. And to determine the brake power of the engine, a dynamometer was used to measuring the brake torque at certain rpm.

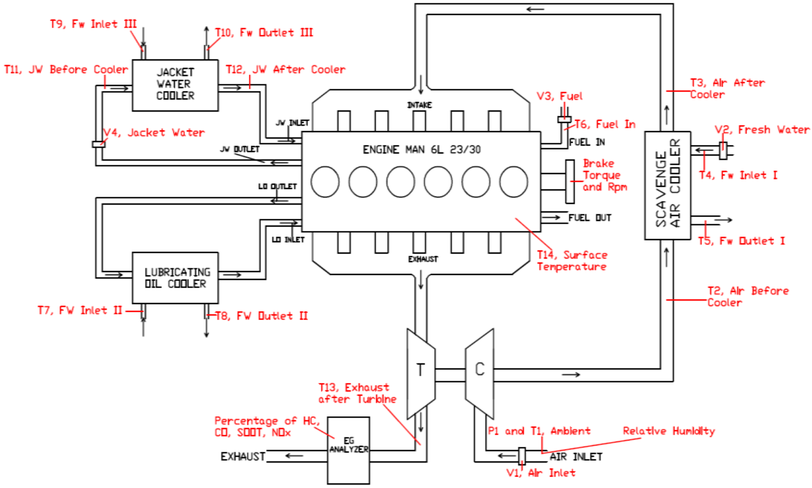


figure III.3 Engine Test Layout

The explanation of the measuring equipment would be discussed in order to give a better understanding of the scheme in figure 3.3. And the following measurement equipment that used for engine test were:

1. Dynamometer

   In order to measure the brake power output, the hydraulic dynamoter Zöllner-Kiel type 9N38/12F was used. It is coupled to the engine and used to measure the brake torque at certain engine rpm. The maximum power and rpm that could be measured are 1200 kW 3500 rpm respectively.

figure III.4 Hydarulic Dynamometer

2. Fuel Oil System

The volumetric flow rates of the fuel oil was measured using the volumetric flowmeter that shown in figure 3.5. And also the temperature of the fuel oil could be measured at the inlet and outlet of the engine.

figure III.5 Volumetric Flow Rates of Fuel Oil

The system diagram of fuel oil system is shown in figure 3.6, which explained the diagram of fuel system which operated with marine diesel oil (MDO). The MDO is stored in the storage tank. Before it transferred to the service tank, the fuel would be cleaned from water and dirt by means of purifier. To ensure the satisfactory suction during start-up, the standby pump is operated to transfer fuel from service tank to engine inlet. While, during service operation, the primary fuel pump (engine driven pump) is used to circulates the fuel within the system.

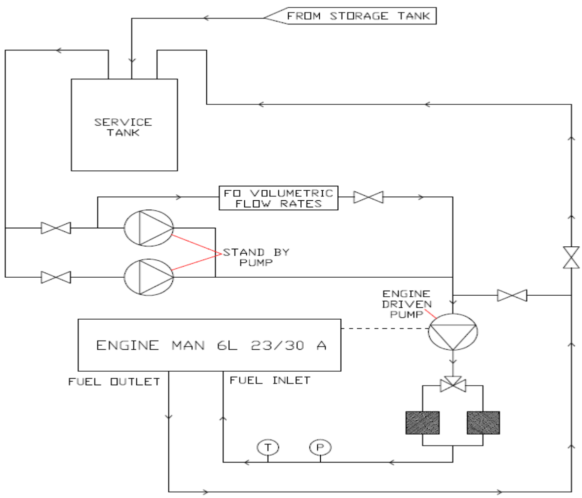


figure III.6 Diagram of Fuel Oil System

3. Cooling System

There are three types of a cooler that work within the engine system which are scavenge air, lubricating oil, and jacket water cooler. The system is divided into two systems which are high temperature cooling system and low temperature cooling system. The layout of the system is shown in figure 3.9 and 3.13. The volumetric flow rates of the coolant of the cooler were measured by flowmeter that installed on its pathways, as shown in figure 3.7. The inlet and outlet temperature also could be measured by thermocouples that installed on the system.

figure III.7 Volumetric Flow Rates of LT Cooling System

figure III.8 Stand By Pump of LT Cooling System

The low temperature (LT) cooling system was designed to cool down the scavenge air, lubricating oil, and jacket water. And after flows out from jacket water cooler, the water from the LT cooling system will flows to the mixing tank and will be cooled down by sea water before its recirculates

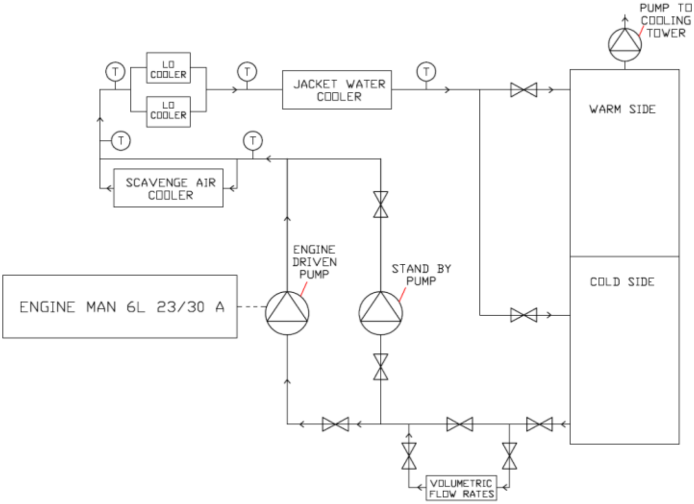


figure III.9 Low Temperature Cooling System

figure III.10 Scavenge Air Cooler

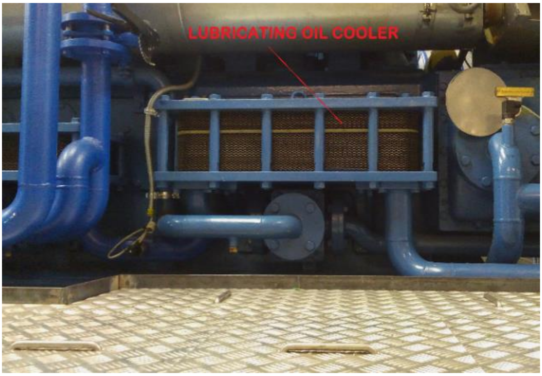


figure III.11 Lubricating Oil Cooler

figure III.12 Jacket Water Cooler

While The high temperature (HT) cooling system works to cooled down the cylinder of the engine. And the water from high-temperature cooling system will be cooled down by jacket water cooler (intersection between HT and LT cooling system). In engine stand-by condition, the jacket water is circulated by means of standby pump, and after engine runs in stable condition, it will be replaced by centrifugal pump which is coupled to the engine.

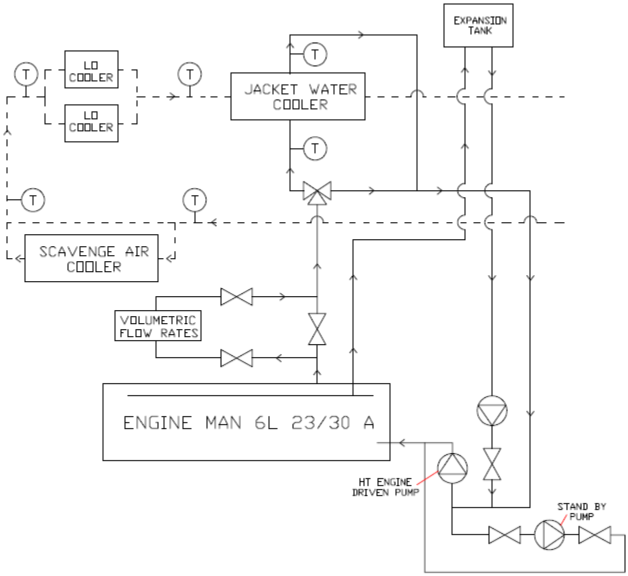


figure III.13 High Temperature Cooling System

4. Charge Air and Exhaust System

The layout system of exhaust gas and charge air is shown in figure 3.14. In order to determine the thermal energy losses inside the exhaust gas, the temperature differences between inlet air and exhaust gas are measured by thermocouples that installed on its pathways. While the mass flow rates of the exhaust gas were determined by the summation of mass flow rates of fuel and inlet air.

The volumetric flow rates of inlet air would be measured by means of flow meter that installed on the inlet air piping system. And the chemical losses would be determined by measuring unburned fuel material and NOx gasses inside the exhaust gas by means of exhaust gas analyzer, shown in figure 3.15.

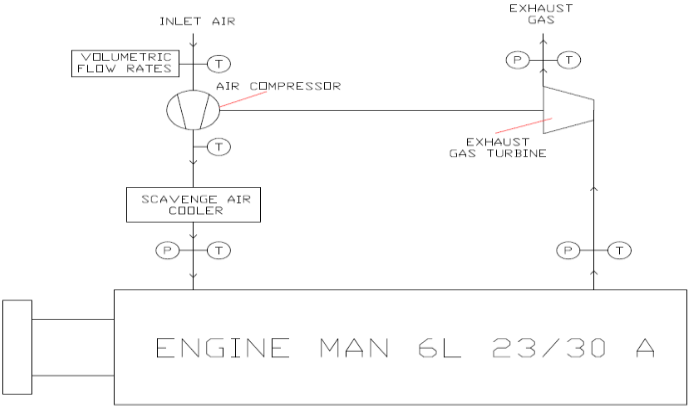


figure III.14 Exhaust and Inlet Air System

figure III.15 Exhaust Gas Analyzer

5. Control Panel and Engine Control Room

All the data and parameters of the engine MAN 6 L23/30A is shown at the control panel in the engine control room (ECR). It was provided with necessary instruments to measure engine parameters. These signals are interfaced to computer through high speed data acquisition. And also the parameters can be seen on the local instrument that installed on the engine system. The control panel on the ECR shows the all the parameters of each engine systems (fuel oil, HT cooling, LT cooling, exhaust, etc.) and also a torque measurement that measured by hydraulic dynamometer.

figure III.16 Control Panel of Engine Control Room

CHAPTER IV
RESULTS AND DISCUSSION

On this chapter would explain the results of calculation which was made from the MAN 6L 23/30A engine test. The data was collected from 50%-90% engine's load. All of the collected data was used to calculates the energy balance and would indicates how good the energy distribution inside the engine. And also the example of calculation of energy efficiency balance would be explained in subcontents 4.1.

4.1 Example of Calculation

The required data which obtained from the engine test is shown below:

Table IV.1 Data from 85% Engine Test

NO	PARAMETER	VALUE	UNITS
AIR PROPERTIES			
1	Ambient temperature	23,00	°C
2	Ambient Air Pressure	1018,10	mbar
3	Humidity	35,1%	
FUEL OIL			
1	Fuel Temperature	21,00	°C
2	Fuel Density @ reference Temp	0,8360	gr/cm3
3	Coeff. Thermal Exp (α)	0,00067	gr/(cm3 x K)
4	FO Volumetric Flow Rates	169,00	kg / hour
5	Fuel Oil Temp @ Operation	35,60	°C
TORQUE AND RPM			
1	Brake Torque	9020,00	N-m
2	Engine Rpm	853,50	rpm
JACKET WATER SYSTEM			
1	JW inlet temp	74,0	°C
2	JW outlet temp	79,30	°C
3	Average Temperature	76,65	°C

4	Volumetric Flow rates	42,60	m3/hour
SCAVENGE AIR SYSTEM			
1	Temp before cooler	137,6	°C
2	Temp after cooler	42,4	°C
3	Air Pressure after cooler	1,5	bar
4	Volumetric flow rates	4304,4	m3/hour
LUBRICATING OIL SYSTEM			
1	Coolant temp before H/E	36,90	°C
2	Coolant temp after H/E	41,97	°C
3	Average Temperature	39,44	°C
4	Volumetric flow rates	27,50	m3/hour
EXHAUST GAS SYSTEM			
1	Volumetric flow rates of Air	4304,4	m3/hour
2	Exhaust Gas Temp After Turbine	345,00	°C
3	Average Temperature	280,6	°C
4	NO gases compostition in EG	825,37	ppm (mg / kg EG)
5	NO2 gases composition on EG	75,66	ppm (mg / kg EG)
6	CO gases composition on EG	54,42	ppm (mg / kg EG)
7	HC composition on EG	50,75	ppm (mg / kg EG)
8	Carbon cocentration	7	gr / m3
SURFACE TEMPERATURE			
1	Engine block surface temp	90,4	°C

The detailed calculation of energy balance of diesel engine would be explained in this chapter. The calculation was made based on the data from the engine test at 85% load. The calculation would analyze all the energy distribution, started with energy input from fuel until several energy output for example like shaft power output, heat released to cooling system, exhaust gas, and also chemical losses of exhaust gas. The detailed calculation is explained below:

1. **Humid Air Density** (*Handbook of Chemistry and Physics*, Weast, 1986; page F-8)

- $\rho\ humid\ air\ \left(\frac{kg}{m3}\right) = 1,229\ x \frac{273,15}{T\ amb}\ x \frac{P\ amb - 0,3783\ Pv}{1,013\ x\ 10^5}$
- **Saturated Vapour Pressure (Psv) =** $610,78\ x\ \left(2,178^{\frac{17,67\ x\ Tamb}{243,5+Tamb}}\right)$

  Saturated Vapour Pressure (Psv) = $610,78\ x\ \left(2,178^{\frac{17,67\ x\ Tamb}{243,5+Tamb}}\right) = 2806,59\ Pa$
- **Partial Vapour Pressure (Pv) = Psv x Relative Humidity**
  Partial Vapour Pressure (Pv) = 2806,59 x 35,1% = 985,11 Pa
- $\rho\ humid\ air\ \left(\frac{kg}{m3}\right) = 1,229\ x \frac{273,15}{296,15}\ x \frac{1018,1 - 0,3783\ 985,11}{1,013\ x\ 10^5} = 1,1938\ kg/m3$

2. **Lower Heating Value of Fuel Oil** *(EN ISO 3675)*

- $LHV\ (\frac{kJ}{kg}) = 4,19\ x\ (11040 - (2180\ x\ \rho\ fuel\ at\ 15°C) + 880\ x\ \rho\ fuel\ at\ 15°C)$
- $\rho\ 15°C = \rho\ @\ reference\ Temp + (\alpha\ x\ (T\ reference - 15))$

  $\rho\ 15°C = 0,8360 + \left(0,00067\ x\ (21 - 15)\right) = 0,84002\ \frac{gr}{cm3}$
- $LHV\ (\frac{kJ}{kg}) = 4,19\ x\ (11040 - \left(2180\ x\ 0,84002\right) + 880\ x\ 0,84002 = 42.909,523\ \frac{kj}{kg}$

3. **Fuel Energy Input into the Engine**

- $Qin\ \left(\frac{kJ}{s}\right) = \dot{m}f\ x\ LHV$
- $Qin\ \left(\frac{kJ}{s}\right) = 169\ \frac{kg}{hour}\ x\ 42707\ \frac{kJ}{kg} = 2004,86\ \frac{kJ}{s}$

4. **Brake Power Output**

- $Pb\ (\frac{kJ}{s}) = \tau b\ x\ \omega$
- $Pb\ \left(\frac{kJ}{s}\right) = 9020\ N - m\ x\ 89,414\ rad/s = 806,52\ \frac{kJ}{s}$

5. Heat Released to Jacket Water

- $Qc\ (\frac{kJ}{s}) = vc\ x\rho c\ x\ Cpc\ x\ (Tc\ out - Tc\ in)$
- $Qc\ \left(\frac{kJ}{s}\right) = \frac{42{,}6}{3600}\frac{m3}{s}\ x\ 974{,}04\frac{kg}{m3}\ x\ 4{,}1953\frac{kJ}{kg\ x\ K}\ x\ (79{,}30°C - 74°C) = \mathbf{280{,}463}\frac{kJ}{s}$

6. Heat Released to Lubricating Oil

- $Q\ oil\ (\frac{kJ}{s}) = v\ Coolant\ x\ \rho\ Coolant\ x\ Cp\ coolant\ x\ (T\ Coolant\ out - T\ Coolant\ in)$
- $Q\ oil\ \left(\frac{kJ}{s}\right) = \frac{27{,}5}{3600}\frac{m3}{s}\ x\ 990{,}7\frac{kg}{m3}\ x\ 4{,}1805\frac{kJ}{kg\ x\ K}\ x\ (46{,}63°C - 41{,}2°C) = \mathbf{171{,}791}\frac{kJ}{s}$

7. Heat Released to Scavenge Air

- $Qa\ (\frac{kJ}{s}) = va\ x\ \rho\ air\ x\ Cpa\ x\ (T\ air\ after\ cooler - T\ air\ before\ cooler)$
- $Qa\ \left(\frac{kJ}{s}\right) = \frac{4304{,}4}{3600}\frac{m3}{s}\ x\ 1{,}938\frac{kg}{m3}\ x\ 1{,}0097\frac{kJ}{kg\ x\ K}\ x\ (137{,}6°C - 42{,}4°C) = \mathbf{137{,}208}\frac{kJ}{s}$

8. Heat Released to Ambient Air

- $Q\ conv\ \left(\frac{kJ}{s}\right) = \frac{Tsurface - T\ amb}{Surface\ Area\ x\ h\ total}$
- $Q\ conv\ \left(\frac{kJ}{s}\right) = \frac{90{,}4°C - 23{,}0°C}{15{,}582\ m2\ x\ 14{,}582\frac{W}{m2\ x\ K}} = \mathbf{17{,}791}\frac{kJ}{s}$

9. Specific Heat of Exhaust Gas *(Pricinples Calculation of Energy Balance, Hochschule Wismar)*

- $Cp\ Exh = \frac{\dot{m}Exh\ min\ x\ Cp\ Exh\ |_T + \dot{m}Air\ min\ x\ (\lambda - 1)x\ Cp\ air\ |_T + \lambda\ x\ XL\ x\ \dot{m}Air\ min\ x\ Cp\ H2O\ |_T}{1 + \lambda\ (1 + XL)\ \dot{m}Air\ min}$
- $\lambda = \frac{\dot{m}\ Air}{\dot{m}\ Air\ minimum\ x\ \dot{m}\ Fuel}$
    - $\dot{m}\ Air\ min = 14{,}53\frac{kgair}{kg\ fuel} = 2455{,}57\ kg/hour$
    - $\dot{m}\ Exh\ Gas\ min = (14{,}53 + 1)\frac{kg\ EG}{kg\ fuel} = 2624{,}57\ kg/hour$

$$\lambda = \frac{1{,}1938\frac{kg}{m3}\ x\ 4784{,}1\frac{kg}{hour}}{14{,}53\frac{kg\ air}{kg\ fuel}\ x\ 169\frac{kg\ fuel}{hour}} = 2{,}0926$$

- $XL = 0,622 \frac{Ps\ x\ \varphi}{P\ amb - Ps\ x\ \varphi}$

The value of Ps and φ was obtained from the following table which depends on the ambient temperature and relative humidity.

Table IV.2 Humidity factor (XL) and Saturated Vapour Pressure (Principles Calculation of Energy Balance, Hochschule Wismar)

TEMPERATURE	0	5	10	19,9	20	23	30	38	40	47	50
Ps (bar)	0,00611	0,00920	0,01228	0,02326	0,02337	0,03441	0,04241	0,06747	0,07374	0,10846	0,12334
HUMIDITY	HUMIDITY FACTOR (XL)										
40%	0,0015	0,00230	0,0031	0,00587	0,0059	0,00868	0,0107	0,01726	0,0189	0,02828	0,0323
42%	0,001584	0,002421	0,003258	0,006175	0,006205	0,009147	0,011278	0,018207	0,01994	0,029871	0,034127
60%	0,0023	0,00345	0,0046	0,00876	0,0088	0,01309	0,0162	0,02628	0,0288	0,04343	0,0497
72%	0,00278	0,00417	0,00556	0,010609	0,01066	0,015822	0,01956	0,031848	0,03492	0,052994	0,06074
80%	0,0031	0,00465	0,0062	0,01184	0,0119	0,01764	0,0218	0,03556	0,039	0,05937	0,0681
93%	0,003555	0,005365	0,007175	0,013783	0,01385	0,02061	0,025505	0,041761	0,045825	0,070245	0,08071
100%	0,0038	0,00575	0,0077	0,01483	0,0149	0,02221	0,0275	0,04510	0,0495	0,07610	0,0875

- $XL = 0,622 \frac{0{,}03441\ x\ 0{,}006334}{1{,}018 - 0{,}03441\ x\ 0{,}006334} = 0{,}000132$

The value of $Cp\ Exh\ \big|_T$, $Cp\ air\ \big|_T$, and $Cp\ H2O\ \big|_T$ was obtained from the table 4.3 which depends on the exhaust gas temperature (average temperature) after turbocharger.

Table IV.3 Specific heat of exhaust gas, air and H20 (Principles Calculation of Energy Balance, Hochschule Wismar)

TEMPERATURE	180	194,5	200	229,06	250	280,6	300	345
CP Exhaust Gas	1,0737	1,075585	1,0763	1,080252	1,0831	1,08738	1,0901	1,09658
Cp Air	1,0104	1,01134	1,0117	1,013734	1,0152	1,01765	1,0192	1,02325
CP H20	1,8885	1,891835	1,8931	1,900249	1,9054	1,91342	1,9185	1,93092

- $Cp\ Exh\ \big|_{280,6} = 1{,}08738 \frac{kJ}{kg\ x\ K}$
- $Cp\ air\ \big|_{280,6} = 1{,}01765 \frac{kJ}{kg\ x\ K}$

- $Cp\ H20\ \Big|_{280,6} = 1{,}9134 \frac{kJ}{kg\ x\ K}$
- $Cp\ Exh = \frac{(2624{,}57\ x\ 1{,}08738)+(2455{,}57\ x\ (2{,}0926-1)x\ 1{,}01765)+(2{,}0926\ x\ 0{,}00132\ x\ 2455{,}57\ x\ 1{,}9134)}{1+2{,}0926\ x\ (1+0{,}000135)\ x\ 2455{,}57} = 1{,}08658 \frac{kJ}{kg\ x\ K}$

10. Thermal Losses of Exhaust Gas*(Pricinples Calculation of Energy Balance, Hochschule Wismar)*

- $Q\ Exh\ Gas\ \left(\frac{kJ}{s}\right) = (\dot{m}\ f + \dot{m}\ Air) x\ Cp\ Exh\ x\ T\ Exh - \dot{m} Air\ x\ Cp\ Air\ x\ T\ Air$
- $Q\ Exh\ Gas\ \left(\frac{kJ}{s}\right) = (169 + 5138{,}64)\ x\ 1{,}08658\ x\ (345) - 5138{,}64\ x\ 1{,}009\ x\ (23) = 524{,}87 \frac{kJ}{s}$

11. Chemical Losses of Exhaust Gas *(Principles Calculation of Energy Balance, Hochschule Wismar)*

A. NOx gases

- $Heat\ losses\ due\ to\ Nox\ gases\ \left(\frac{kJ}{s}\right) = (NOx)\ ppm\ x\ \dot{m}\ Air\ x\ NOx\ enthalpy$
- $Heat\ losses\ due\ to\ Nox\ gases\ \left(\frac{kJ}{s}\right) = 901{,}03\ ppm\ x\ 4304{,}4 \frac{kg}{hour}\ x\ 3{,}0123 \frac{kJ}{gr} = 3{,}874 \frac{kJ}{s}$

B. Hydrocarbon

- $Heat\ losses\ due\ to\ hydrocarbon\ \left(\frac{kJ}{s}\right) = (HC)\ ppm\ x\ \dot{m}\ Air\ x\ HC\ enthalpy$
- $Heat\ losses\ due\ to\ hydrocarbon\ \left(\frac{kJ}{s}\right) = 50{,}03\ ppm\ x\ 4304{,}4 \frac{kg}{hour}\ x\ 42{,}707 \frac{kJ}{gr} = 3{,}0937 \frac{kJ}{s}$

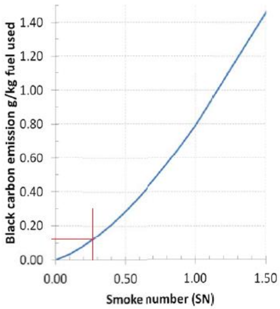
C. CO gases *(Pricinples Calculation of Energy Balance, Hochschule Wismar)*

- $Heat\ losses\ due\ to\ CO\ gases\ \left(\frac{kJ}{s}\right) = 22{,}57\ x\ \left(\frac{131300}{LHV} - 1\right) x\ \left(\frac{\%\ CO}{\%\ CO2+\%CO}\right) x\ \left(\frac{Q\ in}{100}\right)$
- $Heat\ losses\ due\ to\ CO\ gases\ \left(\frac{kJ}{s}\right) = 22{,}57\ x\ \left(\frac{131300}{42707} - 1\right) x\ \left(\frac{0{,}0054\%}{6\%+0{,}054\%}\right) x\ \left(\frac{2004{,}86}{100}\right) = 0{,}851 \frac{kJ}{s}$

D. Soot (*Background Information on Black Carbon Emissions from Large Marine and Stationary Diesel Engines – Definition, Measurement Methods, Emission Factors, and Abatement Technologies, CIMAC, 2012)*

- $\boldsymbol{Soot\ Heat\ Losses = \left(\frac{kj}{s}\right)\ Fuel\ Carbon\ Emission\ x\ C\ enthalpy}$

It is necessary to use graph 2 in order to determine the specific fuel carbon emission (in gr Carbon / kg fuel being used). The value is depends on the smoke number of exhaust gas. In this case the smoke number was 0,27 and the steps to determine the fuel carbon emission is shown below:



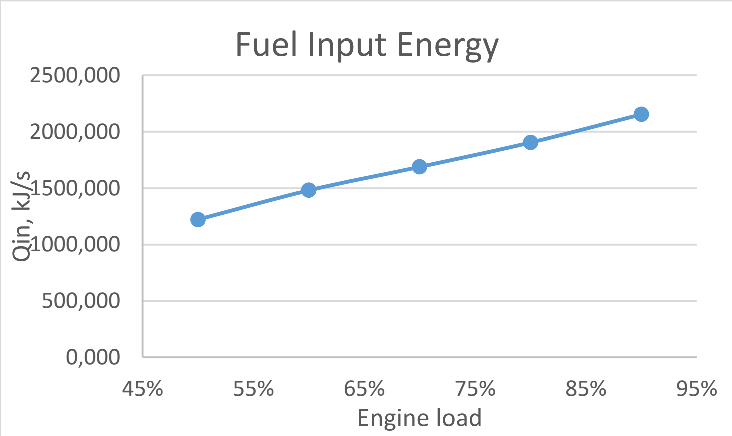
Graph IV.1 Specific Black Carbon Emission (CIMAC, 2012)

From the graph above, the number of specific fuel carbon emission is 0,13 gr Carbon / kg Fuel being used.

- $\boldsymbol{Specific\ Fuel\ Carbon\ Emission = 0,13\,\frac{gr\ Carbon}{kg\ Fuel}}$
- $\boldsymbol{Fuel\ Carbon\ Emission = 0,13\,\frac{gr\ Carbon}{kg\ Fuel}\ x\ 169\,\frac{kgFuel}{hour} =}$
  $\boldsymbol{21,97\,\frac{gr\ Carbon}{hour}}$
- $\boldsymbol{Heat\ Losses\ due\ to\ Soot = Fuel\ Carbon\ Emission\ x\ C\ Enthalpy}$
- $\boldsymbol{Heat\ Losses\ due\ to\ Soot = 21,97\,\frac{grCarbon}{hour}\ x\,\frac{1}{3600}\ x\ 0,1414\,\frac{kJ}{gr} =}$
  $\boldsymbol{0,0009\,\frac{kJ}{s}}$

4.2 Input energy goes to the engine

The results of calculation of input energy goes to the engine is shown at graph 4.2. The results shows the input energy in kJ/s at 50% until 90% engine load.



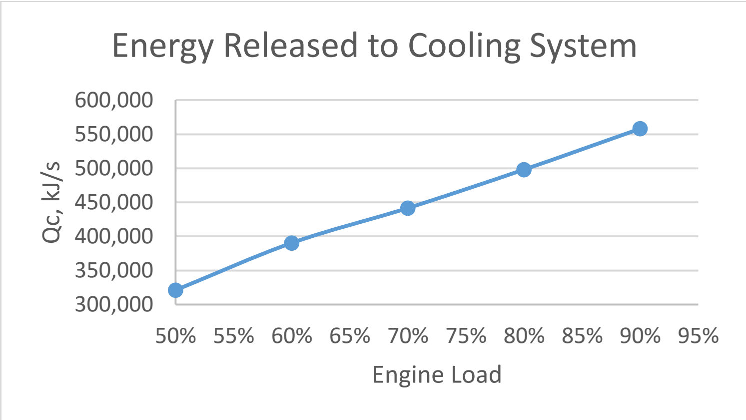
Graph IV.2 Fuel Input Energy

Graph 4.2 shows the input energy goes into the engine. The highest energy input was occurred at 90% load which indicates the value of 2153,15 kJ/s with mass flow rate of fuel of 181,50 kg/hour. The graph trend shows as engine load went higher, the fuel input energy also became higher. It was due to the mass flow rates of the fuel which also increased and the engine would require a higher amount of the fuel to produce a higher rated power

4.3 Heat released to cooling system

The results of the calculation of heat released to cooling system is shown in graph 4.3 and graph 4.4. The results of calculation was made from 3 main components of cooling system which are jacket water cooler, lubricating oil cooler, and scavenge air cooler. It shows the heat released to cooling system at 50% until 90% engine load.

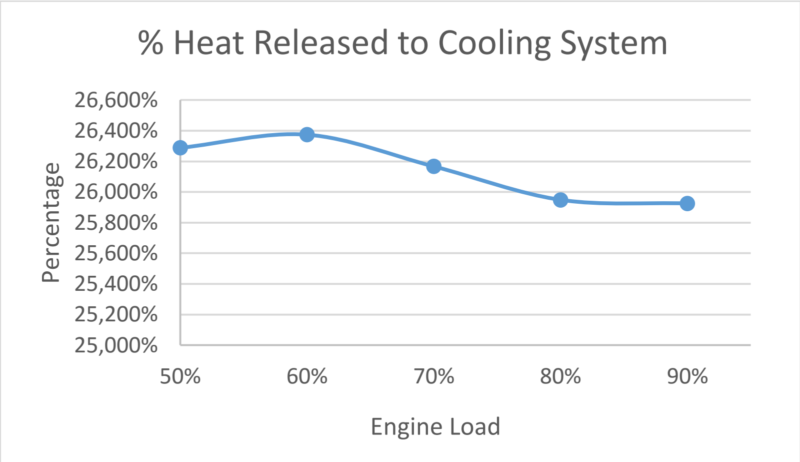
Graph 4.3 shows the total heat released to cooling system. It can be seen as the engine load went higher, the heat released to cooling system also became higher. It was due to the higher mass flow rates of the coolant and also, at high engine load operation, there was a more homogenous temperature distribution throughout combustion chamber and makes a higher combustion temperature and more complete combustion inside the cylinder. This phenomenon gives a results to more heat transfer to the cylinder walls and a fewer uncombusted products carried out to exhaust gas (Wallace, 2007)



Graph IV.3 Total Heat Released to Cooling system

Graph 4.4 shows the percentage of heat released to the cooling system. The percentage means the ratio between heat released to cooling system and energy input goes to the engine. The highest percentage of heat released to cooling system has happened at 60% load, which indicates the value of 26,374%.

The graph has a trend that the percentage of energy released to cooling system reduced as the engine load went higher. Low load operations of diesel engines cause a lower cylinder pressure and thus, lower combustion temperature occurred. The results of low temperature might lead to an ignition problem and a poor combustion which causes an increased amount of unburned fuel inside the cylinder. Thus, this phenomenon leads to a lower efficiency of the engine. And instead distributed to the shaft (as useful work), there was more energy which distributed to the cooling system and exhaust gas.

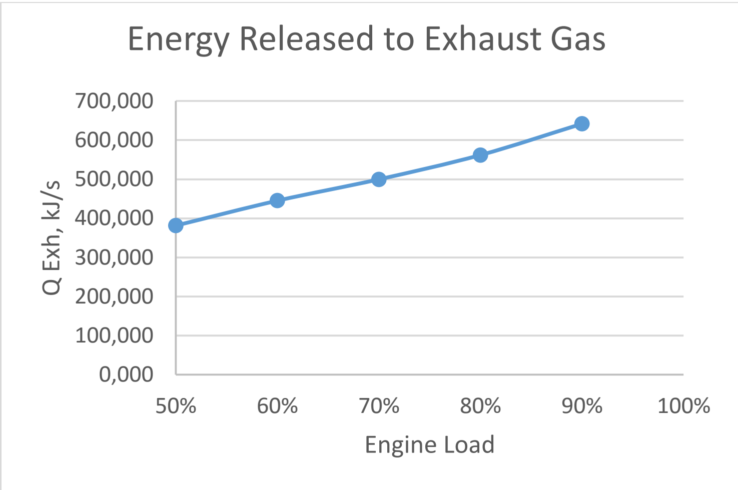


Graph IV.4 Percentage of heat released to cooling system

4.4 Heat Released to Exhaust Gas

The results of calculation of heat released to exhaust gas are shown in graph 4.5 and graph 4.6. Graph 4.5 shows the total energy released to exhaust gas (in kJ/s), while graph 4.6 shows the percentage of heat released to exhaust gas at each engine load. The results calculate how much energy is distributed as heat to the exhaust gas by measuring the mass flow rates of exhaust gas and exhaust temperature after turbocharger. It shows the thermal energy losses of exhaust gas at 50%-90% engine load.

Graph 4.5 shows the total energy that released to exhaust gas at 50% until 90% engine load. The graph shows that a higher engine load will give a result to a higher energy transferred to the exhaust gas. It was due to the higher exhaust gas mass flow rates. In high engine operation, there will be a higher amount of fuel injected into engine's cylinders. And to maintain the air-fuel ratio in relatively constant value, the amount of inlet air also needs to be increased. Since mass flow rates of exhaust gas equal to the summation of mass flow rates of fuel and air, its value also would be increased as well.

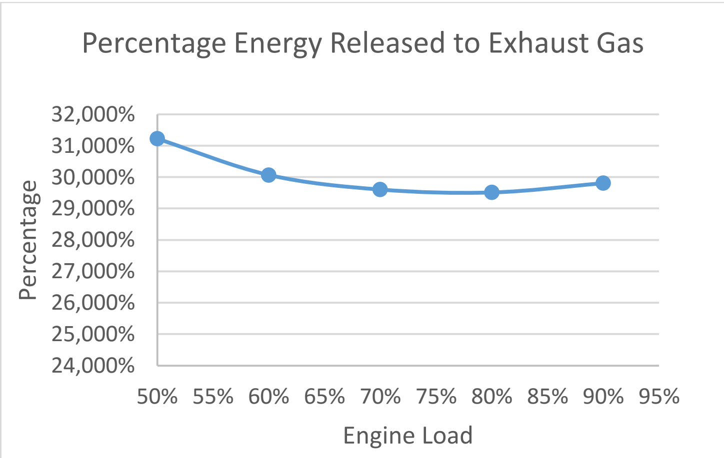


Graph IV.5 Energy released to exhaust gas

Graph 4.6 shows the percentage of energy which released to exhaust gas at 50% until 90% engine load. It can be seen from the graph 7 that the energy released to exhaust stay on the relatively constant fluctuation which varies from 29% until 31% of the total energy.

The high load operation has slightly lower percentage energy released to the exhaust gas. It might be explained because the highest turbocharger efficiency was occured, typically, at 80-85% engine load. And it can be seen from the graph that the lowest heat released of exhaust was occurred at 80% engine load. Thus, this phenomenon would produce a higher engine efficiency and useful work transferred to the engine's shaft and, moreover, less wasted energy transferred to the exhaust gas.

At high engine load, it also had a slightly lower exhaust gas temperature after turbocharger. It might be explained because, at high engine load, there was a higher mass flow of exhaust gas that flow into exhaust gas turbine. And it means there was a higher energy that utilized by the exhaust gas turbine instead of released to exhaust gas. And it gave a results to a lower exhaust temperature after turbocharger.



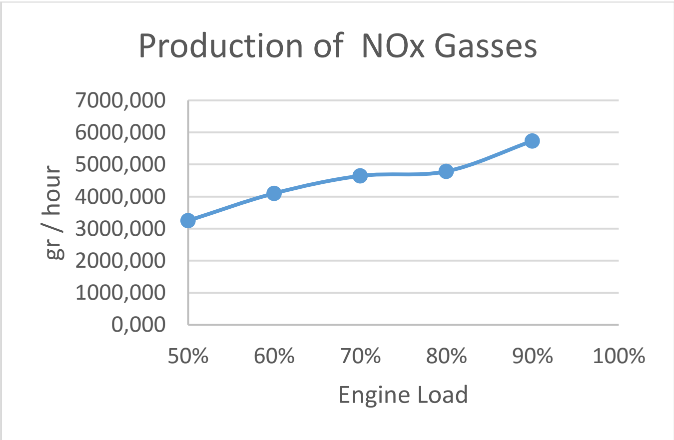
Graph IV.6 Precentage of energy released to exhaust gas

4.5 Chemical Losses of Exhaust Gas

This part will explain the results of chemical losses which obtained by analyzing the combustion products inside the exhaust gas. The substances consist of NOx gasses, Hydrocarbon, CO gasses, and soot. The calculation was made to determine how much the energy lost due to the formation of those substances. And the graphs will show the total production and also the percentage of energy lost due to the formation of those substances. The percentage means the ratio between energy lost due to the substances compares to the fuel input energy.

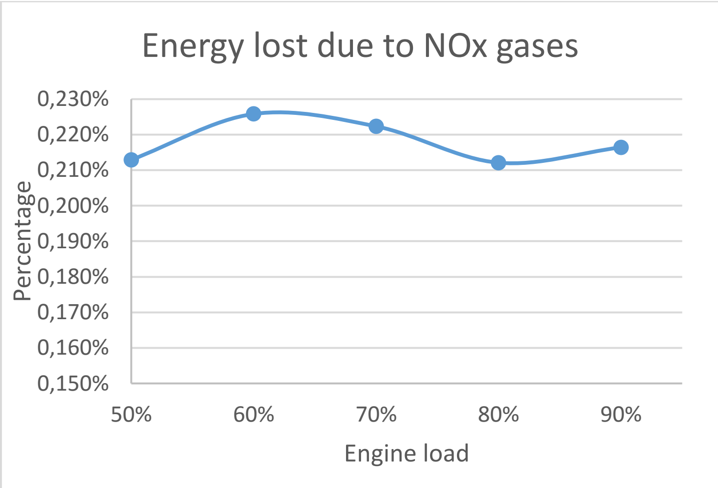
4.5.1 NOx Gases

Graph 4.7 and graph 4.8 shows the total energy and percentage of energy lost due to the formation of NOx gasses. In this case, NOx gasses was the summation of NO_2 and NO gasses. But almost 95% of the composition was NO gasses.



Graph IV.7 Production of NOx gases

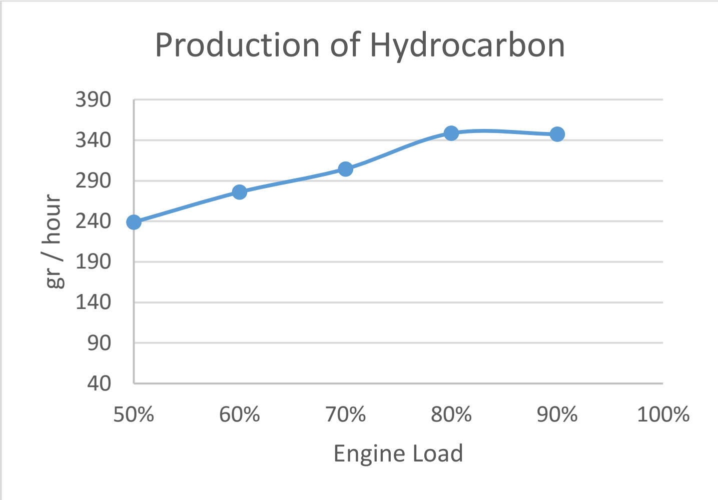
Graph 8 shows the total production of NOx gasses. It can be seen from the graph as the engine load increased, the production also became increased. It might be explained because a high engine load operation tends to have a higher combustion temperature. As we know, that NOx gasses is produced by the chemical reaction between nitrogen and oxygen at high temperature.



Graph IV.8 Energy lost due to NOx gases

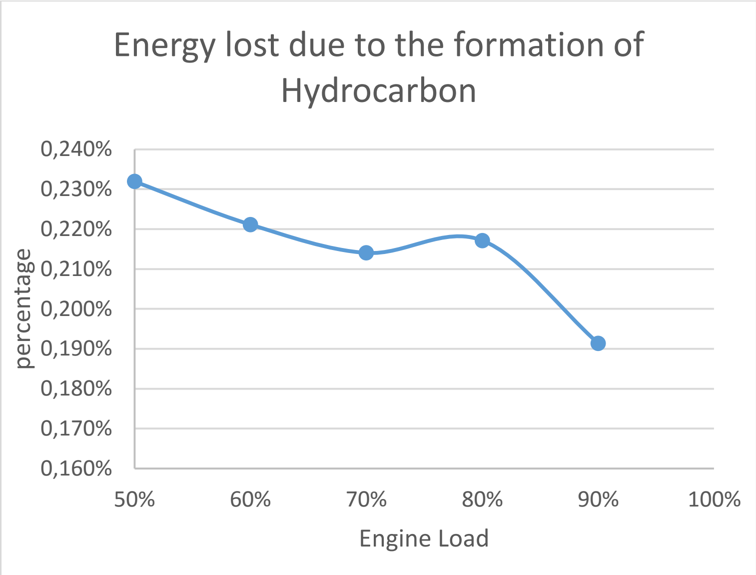
4.5.2 Hydrocarbon

Graph 4.9 and graph 4.10 shows the total production and percentage of energy lost due to the unburned fuel material that analyzed by measuring hydrocarbon inside the exhaust gas. Graph 4.9 shows that as the load became higher the production of hydrocarbon was increased. Even though the composition of hydrocarbon inside the exhaust gas (in units of ppm) was reduced as the engine load went higher, but, the production of hydrocarbon (in gr/hour) went higher due to the increased mass flow rates of the fuel.



Graph IV.9 Production of hydrocarbon

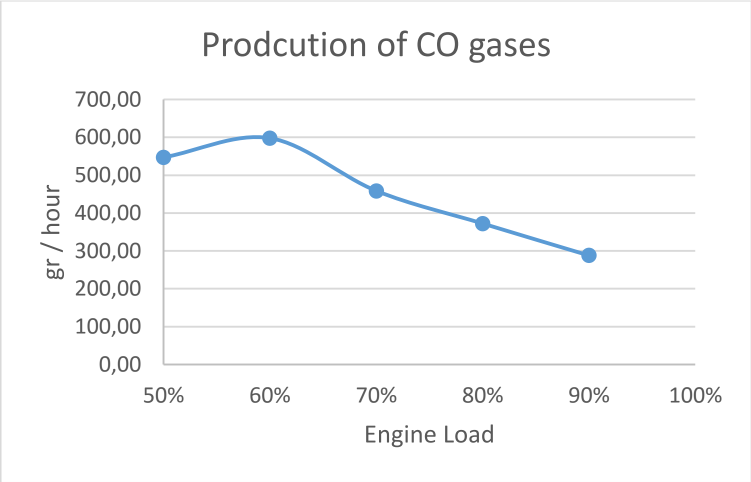
Graph 11 shows the percentage of energy lost due to the formation of hydrocarbon. Even though the production of hydrocarbon increased as the engine load went higher, the opposite things have happened to the energy lost. Because the total energy lost due to the hydrocarbon is relatively small compares to the total energy input from fuel which increased as the engine load went higher. And that is the reason why the percentage of energy due to the formation of hydrocarbon was decreased.



Graph IV.10 Energy losses due to the formation of hydrocarbon

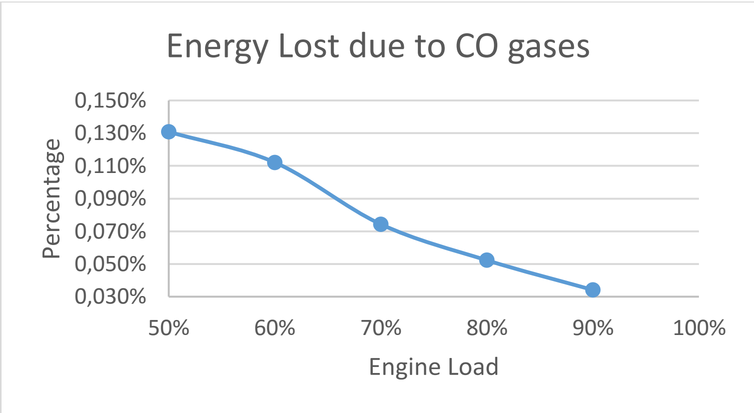
4.5.3 CO gases

Graph 4.11 and graph 4.12 show the total production of CO gases and the percentage of energy lost due to the formation of CO gases respectively. Graph 4.11 shows that as the load became higher the production of CO gases was reduced. It might be explained because, at low engine load, there was no enough oxygen to make a complete oxidation process of carbon. So instead of producing C02 gases, it produced CO gases.



Graph IV.11 Production of CO gases

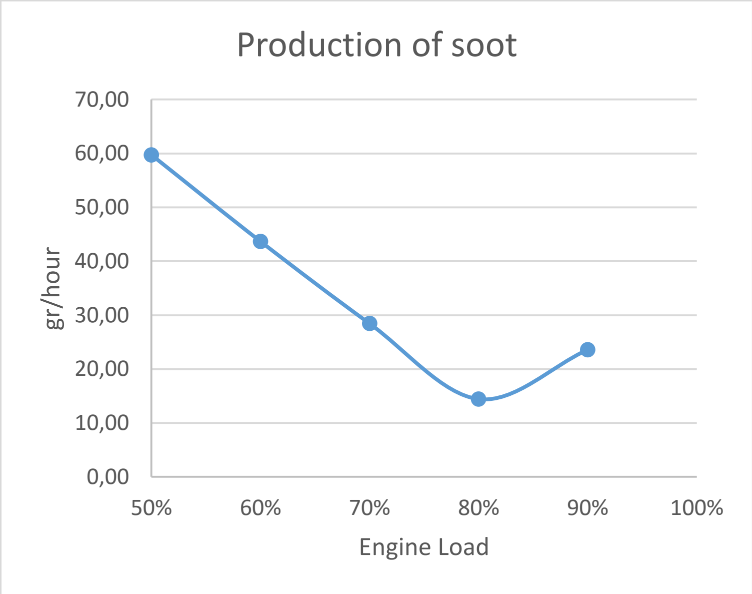
Graph 4.12 shows the percentage of energy lost due to the formation of CO gases. It shows that as the engine load became higher the percentage was reduced. It was due to the lower production of CO gases at higher engine load.



Graph IV.12 Energy lost due to CO gases

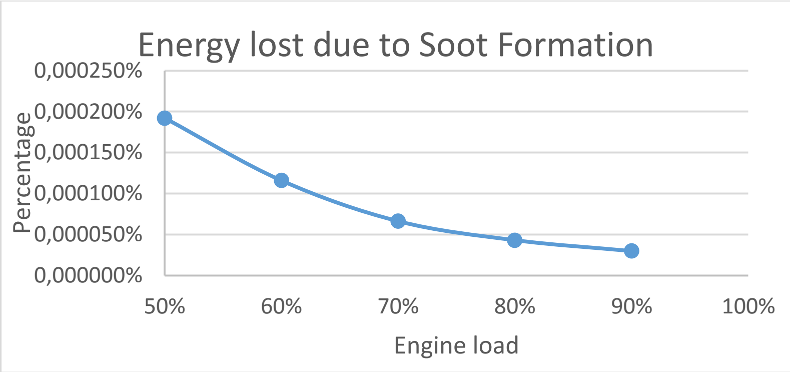
4.5.4 Soot

Graph 4.13 shows the production of soot at 50%-90% engine load. It was determined by measuring the smoke number at each engine load. As the engine load went higher, it produced a lower soot production. It might be explained due to the higher combustion temperature at higher engine load. Thus, the fuel is burned more completely and soot production reduced.



Graph IV.13 Production of soot

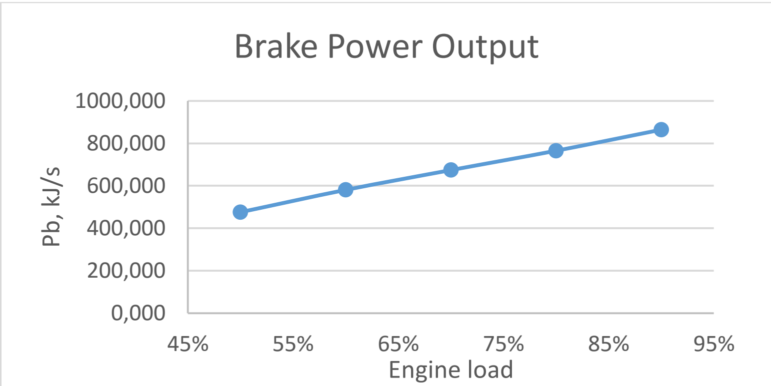
Graph 4.14 shows the percentage of energy losses due to the formation of soot. The graph shows that the higher the engine load, the energy losses would be reduced. It was due to a lower soot production at high engine load.



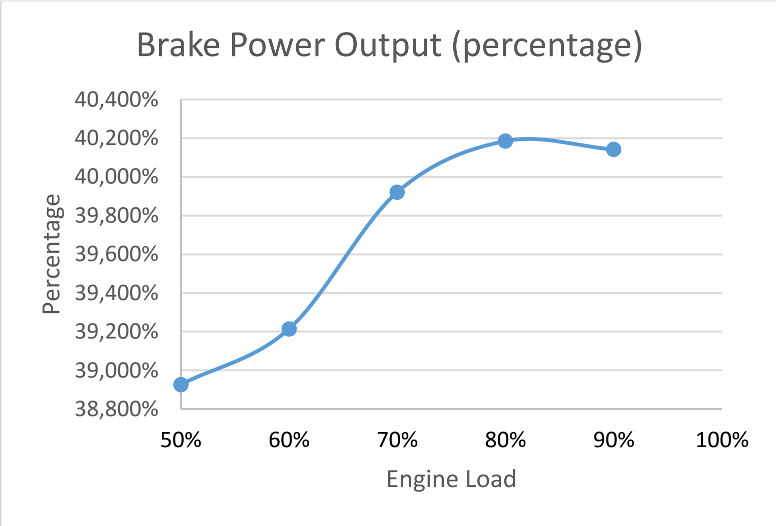
Graph IV.14 Energy Lost due to Soot Formation

4.6 Brake Power Ouput

This part will explain the result of brake power output at certain engine load. The result was made from the calculation which determined by measuring the brake torque and engine rpm. The results are shown in graph 4.15 and graph 4.16. Graph 4.15 shows how much the energy (in kJ/s) that converts into useful work at certain engine load. While graph 4.16 shows the percentage of energy that converts into useful work.



Graph IV.15 Brake power output



Graph IV.16 Percentage of brake power output

Graph 4.16 shows that the thermal efficiency at each engine load. Thermal efficiency was increasing from 50% load until it reached its maximum value at 80% load which indicates the value of 40,185%. It might be explained due to the higher combustion pressure, increased turbulence of air-fuel mixture and the fuel was injected into the cylinder at higher speed, thus, diesel droplets got enough time to make a complete combustion process and resulting in higher conversion of chemical energy (from fuel) into mechanical energy.

But, as the load increased further (to 90% load), the thermal efficiency has reduced. The first reason might be explained due to the lower efficiency of turbocharger. Because, typically, the highest turbocharger efficiency occurred at 80-85% engine load. It means that there was a higher energy that wasted to the exhaust funnel rather than converted into useful work. It also can be seen at graph 4.6 that at 90% engine load there was a slightly higher energy wasted to the exhaust gas than at 80% engine load.

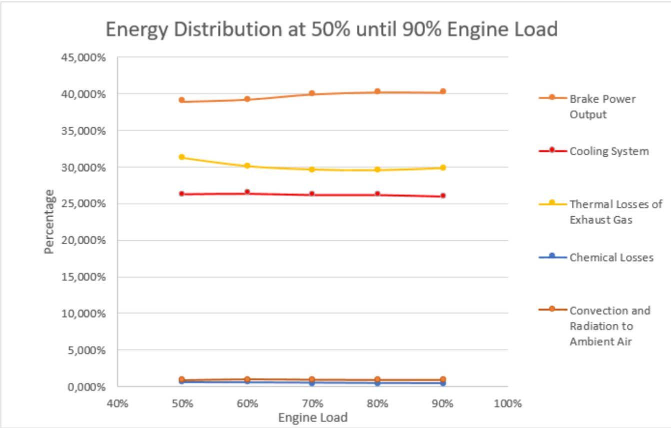
The other reason is the combustion duration. As we know that the engine would require a higher amount of the fuel at higher engine load. And It would increase the duration of injection. This phenomenon would reduce the duration of combustion, so the fuel did not have enough time to make a complete combustion. Thus, the complete conversion of chemical energy (from fuel) into mechanical energy did not take place and led to a lower efficiency.

4.7 Energy Distribution

This part shows the energy distribution of certain engine load. It would explain how much the energy distribution from fuel to the several output for example, to the shaft power output, cooling system, exhaust gas, and also chemical losses. The energy balance was measured at 50% until 90% engine load. The results are shown in table 4.1.

Unfortunately, the results could not indicate 100% of energy balance. There were unaccounted losses that seem hard to measure it in real conditions. The accounted losses could depend on the correctness of the measuring instrument, heat transfer to surroundings, operating auxiliary equipment and also approximation formula that used in the calculation. But, it was set that the maximum unaccounted losses were 3%. Thus, if the total energy output indicates less than 97% of the total energy input, it could not be considered as a correct calculation.

The results of energy distributions can be seen in graph 4.17 and table 4.1. The results show the energy balance which consists of brake power output, thermal losses of exhaust gas, heat released to the cooling system, chemical losses (due to NOx, CO gases, hydrocarbon, and soot), and heat released to the ambient air by convection and radiation. The average of energy balance indicates the value of 97,343%. The highest energy balance occurred at 50% engine load with the value of 97,90%. And the lowest value occurred at 70% engine load which indicates the value of 97,15%.



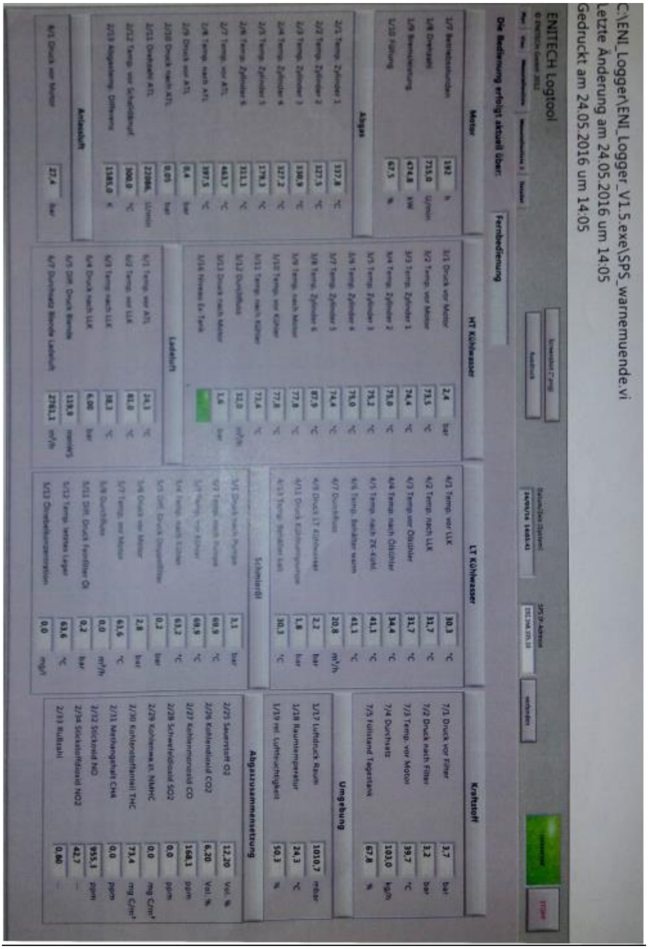
Graph IV.17 Energy Distribution from 50% until 90% engine load

Table IV.4 Energy Balance from 50% until 90% engine load

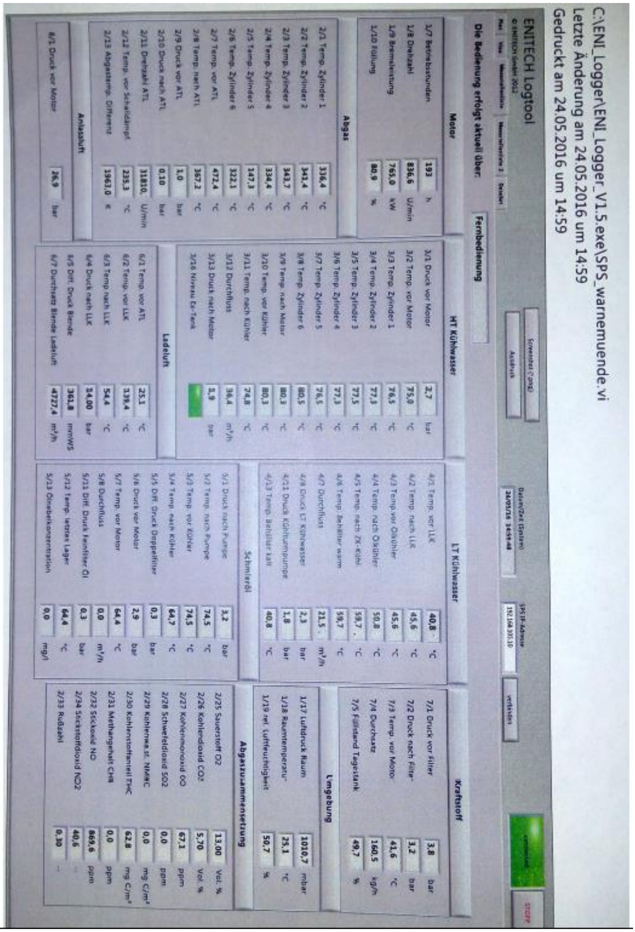
NO	PARAMETERS	50%		60%		70%		80%		90%	
		P (kJ / s)	Percentage	P (kJ / s)	Percentage	P (kJ / s)	Percentage	P (kJ / s)	Percentage	P (kJ / s)	Percentage
1	Fuel Input Energy	1221,895		1480,509		1688,113		1904,020		2153,145	
2	Brake Power Output	475,645	38,927%	580,572	39,214%	673,915	39,921%	765,130	40,185%	864,307	40,142%
3	Heat Released to Cooling System										
	3.1 Heat Released to Jacket Water	156,258	12,788%	200,322	13,531%	203,901	12,079%	231,275	12,147%	230,941	10,726%
	3.2 Heat Released to Scavenge Air	41,262	3,377%	72,394	4,890%	104,105	6,167%	126,183	6,627%	179,186	8,322%
	3.3 Heat Released to Lubricating Oil	123,691	10,123%	117,755	7,954%	133,723	7,921%	140,746	7,392%	148,083	6,878%
	3.4 Sea Water Side	315,378	25,811%	380,969	26,374%	449,155	26,607%	480,649	25,244%	541,658	25,157%
4	Thermal Losses of Exhaust Gas	381,519	31,224%	445,165	30,068%	499,748	29,604%	561,862	29,509%	641,777	29,807%
5	Heat Released to Ambient Air	10,822	0,886%	14,703	0,993%	15,917	0,943%	17,395	0,914%	19,586	0,910%
6	Chemical losses										
	8.1 NO Gases	2,602	0,213%	3,312	0,224%	3,753	0,222%	3,827	0,201%	4,634	0,215%
	8.2 NO2 Gases	0,028	0,002%	0,029	0,002%	0,033	0,002%	0,044	0,002%	0,041	0,002%
	8.3 CO Gases	1,598	0,131%	1,660	0,112%	1,255	0,074%	0,996	0,052%	0,736	0,034%
	8.4 HC	2,834	0,232%	3,244	0,219%	3,613	0,214%	3,918	0,206%	4,098	0,190%
	8.5 SOOT	0,00235	0,00019%	0,002	0,00012%	0,00112	0,00007%	0,00057	0,00003%	0,00093	0,00004%
7	Uncounted Losses	25,632	2,098%	41,353	2,793%	48,149	2,852%	52,644	2,765%	59,755	2,775%
8	Brake Specific Fuel Oil Consumption	0,217	kg/kwH	0,215	kg/kwH	0,211	kg/kwH	0,20977	kg/kwH	0,2100	kg/kwH
	TOTAL	1196,26	97,90%	1439,16	97,21%	1639,96	97,15%	1851,38	97,24%	2093,39	97,22%

APPENDIX

1. Data from Engine Test at 50% Load



2. Data from engine test at 80% load



3. Data from engine test at 60%, 70%, and 90% engine load

				Load	60,42%	70,15%	89,91%
Nr			**Messgröße**	**Einheit**	11:17	15:15	11:04
1	m		Umgebungsluftdruck	kPa	1005,6	1016,5	1025,9
2	m		Luftfeuchte	%	59,4	34	35,8
3	m		Raumtemp.	°C	24,8	24,7	23,8
4		1/8	Motordrehzahl	min^{-1}	802	800,6	868,9
5		1/9	Bremmsleistung	kW	580	673,4	863,1
6		1/10	Füllung	%	70,7	76,6	85,1
7		2/1	Abgastemperatur Zyl. 1	°C	326,4	333,3	334,6
8		2/2	Abgastemperatur Zyl. 2	°C	309,9	334,7	341,6
9		2/3	Abgastemperatur Zyl. 3	°C	323,4	335	340
10		2/4	Abgastemperatur Zyl. 4	°C	319,1	327,4	331,3
11		2/5	Abgastemperatur Zyl. 5	°C	322,2	319,5	339,1
12		2/6	Abgastemperatur Zyl. 6	°C	305,1	313,6	313,9
13			**Abgastemperaturen Durchschnitt**	**°C**	317,68	**327,25**	**333,42**
14		2/7	Abgastemp. Vor ATL	°C	449	465,6	460,1
15		2/8	Abgastemp. nach ATL	°C	371,3	323,3	341,3
16	m	2/9	Druck vor ATL	bar	0,6	0,8	1,3
17		2/10	Druck nach ATL	bar	0,1	0,1	0,1
18		2/11	Drehzahl ATL	min^{-1}	26706	28906	33856
19		3/3	HT Temp. Zyl. 1	°C	75	74,4	76
20		3/4	HT Temp. Zyl. 2	°C	75,7	75,2	76,8
21		3/5	HT Temp. Zyl. 3	°C	75,8	75,3	76,8
22		3/6	HT Temp. Zyl. 4	°C	75,7	75,2	76,8
23		3/7	HT Temp. Zyl. 5	°C	75	74,4	76
24		3/8	HT Temp. Zyl. 6	°C	114,9	76,3	79,5
25			**HT Temp. Durchschnitt**	**°C**	82,02	**75,13**	**76,98**

26		3/10	HT Temp. vor Kühler	°C	78	73,2	79,5
27		3/11	HT Temp. Nach Kühler	°C	73,6	77,8	74,8
28		3/12	Volumenstrom HT Motor	m³/h	40,1	39,9	43,3
29		4/1	Temperatur vor LLK	°C	30,5	34,8	37,7
30		4/2	Temperatur nach LLK	°C	33,1	38,3	43
31		4/3	LT Temp. vor Ölkühler	°C	33,1	38,3	43
32		4/4	LT Temp. nach Ölkühler	°C	37,2	42,9	47,5
33		4/5	LT Temp. Nach Zentralkühler	°C	44	50,5	54,7
34		4/7	Volumenstrom LT für Ölkuehler	m³/h	24,3	24,7	27,4
35		4/8	Druck Kühlwasser	bar	3	2,9	3,3
36		5/1	Öldruck nach Pumpe	bar	3,4	3,1	3,3
37		5/2	Öltemperatur nach Pumpe	°C	69,8	71,9	71,2
38		5/3	Öltemperatur vor Kühler	°C	69,8	71,9	71,2
39		5/4	Öltemperatur nach Kühler	°C	59,2	60,2	61,5
40		5/6	Öldruck vor Mootor	bar	3,1	2,9	2,9
41		5/7	Öltemperatur vor Motor	°C	59,4	59,7	60,8
42		5/8	Volumenstrom Öl	m³/h			
43		6/1	Temperatur vor ATL	°C	24,8	24,7	23,8
44		6/2	Temperatur vor LLK	°C	103,3	125	149
45		6/3	Temperatur nach LLK	°C	40,5	47,6	51,2
46		6/4	Druck nach LLK	bar	0,9	1,1	1,7
47							
48	m	6/7	Volumestrom Ladeluft	m3/h	3521	4058	5450
49		7/4	Brennstoffverbrauch	kg/h	124,8	142,3	181,5
50							
51		2/25	O2 Gehalt Abgas	Vol.%	13	12,9	13,41
52			CO Gehalt Abgas	ppm	144,02	95,4	43,81
53		2/26	CO2 Gehalt Abgas	Vol.%	5,7	5,78	5,4
54		2/27	SO2 Gehalt Abgas	Vol.%			

55		2/29	THC Gehalt Abgas	ppm	66,48	63,35	52,85
56			NO Gehalt	ppm	962,38	932,89	847,24
57			NO2 Gehalt	ppm	34,61	33.55	30,474
58		2/32	NOx Gehalt abgas	ppm	996,99	966,45	877,71
59			Rußzahl		0,58	0,4	0,22

4. Exhaust gas emissions from engine's manual *(MAN B&W)*

Alpha

MAN B&W

Exhaust Gas Emission

Engine		Type	23/30A	28/32A
Power		kW/cyl	160	245
Max combustion pressure		P_{max}	135	145
Speed		RPM	900	775
Exhaust gas quantity normal volume		m^3/kWh	5.82	6.43
O_2, dry 4)		Vol. %	12.4	13.7
NO 1)	15% O_2	ppm, dry	480	700
	15% O_2	g/kWh	5.8	9.2
NO_2 1)	15% O_2	ppm, dry	36	50
	15% O_2	g/kWh	0.4	0.7
NO_x EPA 2)	15% O_2	ppm, dry	516	750
	15% O_2	g/kWh	6.2	9.9
NO_x (TA-Luft) 3)	5% O_2	mg/m^3	2828	4100
CO	15% O_2	ppm, dry	70	115
	15% O_2	g/kWh	0.5	0.9
CO (TA-Luft)	5% O_2	mg/m^3	233	380
HC		ppm, dry	60	130
		g/kWh	0.2	0.5
HC (TA-Luft)	5% O_2	mg/m^3	38	82.8
Smoke value		Bosch	0.15	0.12

Note: 1...4, See page 3
Data refer to ISO condition, Diesel oil operation
Conditions: p = 1.013 bar, t = 0°C
EPA = US Environmental Protection Agency (EPA)
TA : Values according to the German Rule 'TA-Luft'

DEPT/SIGN: KONAM/JGR
STATUS: 1993-01-04

Exhaust Gas Emission

PAGE 2 (3)
Ident. no. 2021035-4

5. Standard enthalpy formation for various compounds (*Thermodynamics, Combustion, and Engines, Milton E.)*

Standard Enthalpy of Formation* for Various Compounds

Compound	ΔH°_f (kJ/mol)	Compound	ΔH°_f (kJ/mol)	Compound	ΔH°_f (kJ/mol)	Compound	ΔH°_f (kJ/mol)
$Ag_2O(s)$	−30.6	$C_2H_5OH(l)$	−277.6	$HCl(g)$	−92.3	$NH_4Cl(s)$	−315.4
$Ag_2S(s)$	−31.8	$C_2H_6(g)$	−84.7	$HF(g)$	−268.6	$NH_4NO_3(s)$	−365.1
$AgBr(s)$	−99.5	$C_3H_8(g)$	−103.8	$HgO(s)$	−90.7	$NiO(s)$	−244.3
$AgCl(s)$	−127.0	$n\text{-}C_4H_{10}(g)$	−124.7	$HgS(s)$	−58.2	$NO(g)$	+90.4
$AgI(s)$	−62.4	$n\text{-}C_5H_{12}(l)$	−173.1	$HI(g)$	+25.9	$NO_2(g)$	+33.9
$Al_2O_3(s)$	−1669.8	$CO(g)$	−110.5	$HNO_3(l)$	−173.2	$Pb_3O_4(s)$	−734.7
$BaCl_2(s)$	−860.1	$CO_2(g)$	−393.5	$KBr(s)$	−392.2	$PbBr_2(s)$	−277.0
$BaCO_3(s)$	−1218.8	$CoO(s)$	−239.3	$KCl(s)$	−435.9	$PbCl_2(s)$	−359.2
$BaO(s)$	−558.1	$Cr_2O_3(s)$	−1128.4	$KClO_3(s)$	−391.4	$PbO(s)$	−217.9
$BaSO_4(s)$	−1465.2	$Cu_2O(s)$	−166.7	$KF(s)$	−562.6	$PbO_2(s)$	−276.6
$Ca(OH)_2(s)$	−986.6	$CuO(s)$	−155.2	$Mg(OH)_2(s)$	−924.7	$PCl_3(g)$	−306.4
$CaCl_2(s)$	−795.0	$CuS(s)$	−48.5	$MgCl_2(s)$	−641.8	$PCl_5(g)$	−398.9
$CaCO_3(s)$	−1207.0	$CuSO_4(s)$	−769.9	$MgCO_3(s)$	−1113	$SiO_2(s)$	−859.4
$CaO(s)$	−635.5	$Fe_2O_3(s)$	−822.2	$MgO(s)$	−601.8	$SnCl_2(s)$	−349.8
$CaSO_4(s)$	−1432.7	$Fe_3O_4(s)$	−1120.9	$MgSO_4(s)$	−1278.2	$SnCl_4(l)$	−545.2
$CCl_4(l)$	−139.5	$H_2O(g)$	−241.8	$MnO(s)$	−384.9	$SnO(s)$	−286.2
$CH_3OH(l)$	−238.6	$H_2O(l)$	−285.8	$MnO_2(s)$	−519.7	$SnO_2(s)$	−580.7
$CH_4(g)$	−74.8	$H_2O_2(l)$	−187.6	$NaCl(s)$	−411.0	$SO_2(g)$	−296.1
$CHCl_3(l)$	−131.8	$H_2S(g)$	−20.1	$NaF(s)$	−569.0	$SO_3(g)$	−395.2
$C_2H_2(g)$	+226.7	$H_2SO_4(l)$	−811.3	$NaOH(s)$	−426.7	$ZnO(s)$	−348.0
$C_2H_4(g)$	+52.3	$HBr(g)$	−36.2	$NH_3(g)$	−46.2	$ZnS(s)$	−202.9

* All standard enthalpy values are at 25°C and 1 atmosphere of pressure.

6. Properties of fresh water *(engineering toolbox.com)*

Temperature - *t* -	Absolute pressure - *p* -	Density - *ρ* -	Specific volume - *v* -	Specific Heat - c_p -	Specific entropy - *e* -
(°C)	(kN/m²)	(kg/m³)	10^{-3} (m³/kg)	(kJ/(kg K))	(kJ/(kg K))
0 (Ice)		916.8			
0.01	0.6	999.8	1.00	4.217	0
4 (maximum density)	0.9	1000.0		4.205	
5	0.9	1000.0	1.00	4.202	0.075
10	1.2	999.8	1.00	4.192	0.150
15	1.7	999.2	1.00	4.1855 1)	0.223
20	2.3	998.3	1.00	4.182	0.296
25	3.2	997.1	1.00	4.180	0.367
30	4.3	995.7	1.00	4.178	0.438
35	5.6	994.1	1.01	4.178	0.505
40	7.7	992.3	1.01	4.179	0.581
45	9.6	990.2	1.01	4.181	0.637
50	12.5	988	1.01	4.182	0.707
55	15.7	986	1.01	4.183	0.767
60	20.0	983	1.02	4.185	0.832
65	25.0	980	1.02	4.188	0.893
70	31.3	978	1.02	4.191	0.966
75	38.6	975	1.03	4.194	1.016
80	47.5	972	1.03	4.198	1.076
85	57.8	968	1.03	4.203	1.134
90	70.0	965	1.04	4.208	1.192
95	84.5	962	1.04	4.213	1.250
100	101.33	958	1.04	4.219	1.307

7. Specific of exhaust gas, air, H20 and water vapour contents *(Principles calculation of energy balance, Hochscule Wismar)*

TEMPERATURE	180	194,5	200	229,06	250	280,6	300	345
CP Exhaust Gas	1,0737	1,075585	1,0763	1,080252	1,0831	1,08738	1,0901	1,09658
Cp Air	1,0104	1,01134	1,0117	1,013734	1,0152	1,01765	1,0192	1,02325
CP H20	1,8885	1,891835	1,8931	1,900249	1,9054	1,91342	1,9185	1,93092

TEMPERATURE	180	194,5	200	229,06	250	280,6	300	345
CP Exhaust Gas	1,0737	1,075585	1,0763	1,080252	1,0831	1,08738	1,0901	1,09658
Cp Air	1,0104	1,01134	1,0117	1,013734	1,0152	1,01765	1,0192	1,02325
CP H20	1,8885	1,891835	1,8931	1,900249	1,9054	1,91342	1,9185	1,93092

CHAPTER V
CONCLUSION AND RECOMMENDATION

This chapter will focus on conlclusions that were made based on the results and analysis that obtained from the MAN 6L 23/30 engine test. This chapter also will give a recommendation regards to determine in which engine load the energy balance would be optimal. Also, the goals and objectives thtat were outlined in chapter 1 will be discussed to show how well the work in this thesis met the expecatations.

The purpose of performing an energy balance study for engine MAN 6L 23/30A was achieved by accomplishing the main tasks: collecting the previous engine's data, developing a calculation program, preparing the laboratory work, running and collecting the required data from the engine test, and making an analysis and report regarding to the energy balance of diesel engine. The details of these steps have been described at the previous chapters and in general they were all have already accomplished

5.1 Energy Distribution Conclusions

The results of the energy balance comparison that were modeled at 50%-90% engine load are shown in graph 3 until graph 18. Graph 18 shows all the energy distribution to the 5 areas (brake power output, cooling system, exhaust gas, chemical losses, and ambient air). The results indicate that the average value of the energy balance was 97,343% of the total energy. It can be concluded that the results were qualified with the minimal value that already set before (97%).

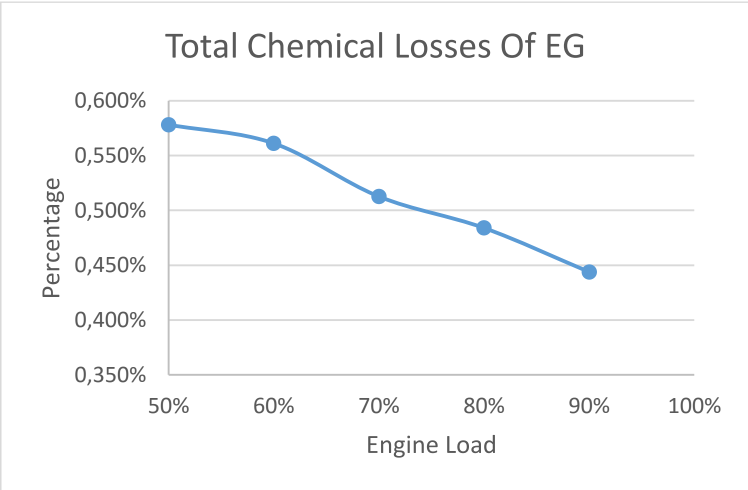
The 100% fuel energy was distributed with the average value of 39,67% , 26,14%, and 30,04% to the brake power output, cooling system, and exhaust gas respectively. While the energy lost distribution to ambient air and chemical losses was found to be quite small compares to the three areas which mentioned before. And the unaccounted indicates an average value of 2,70% of the total energy.

From graph 17 it can be seen that as the engine load goes higher the energy converted into useful work also become increased. It means that the brake thermal efficiency of the engine also increased as the engine load goes higher. The highest percentage of energy that converts into useful work occurred at 80% engine load which indicating the value of 40,185% of the total energy. The phenomenon might be explained due to the increased turbulence of air-fuel mixture when the engine was operated at higher engine load. Thus, it leads to the more homogenous temperature distribution inside the combustion chamber and results in more complete combustion.

Graph 5 and graph 7 shows the percentage of energy released to the cooling system and exhaust gas respectively. The graph has a trend that the percentage of energy released to the cooling system and exhaust gas has reduced as the engine load went higher. It means that, at a higher engine load operation, occurred a more efficient combustion. And there was more energy that converted into useful work rather than released to cooling system or exhaust gas.

Graph 8 until graph 15 shows the production substances and percentage of chemical energy losses. The chemical substances that considered for the calculation are NOx gases, CO gases, hydrocarbon, and soot. The results have a fluctuating trend. For example, NOx gases production would be increased as the engine load went higher due to the high combustion temperature. But, for CO gases, and soot the production of those substances reduced as the engine load went higher.

Thus, in order to make a good conclusion for chemical losses, the summation of energy lost due to the NOx gases, CO gases, hydrocarbon, and soot was made. It is shown in the graph 19. The graph trend shows that as the engine load went higher the energy lost due to chemical losses was reduced. It might be explained because at high engine load operation occurred a more efficient and complete combustion. And it resulted in a fewer uncombusted fuel material inside the exhaust gas.



Graph V.1 Percentage of total chemical losses

5.2 Recommendation for Engine Operation

Every diesel engine operation is always expected to have a high engine efficiency and lower emission. But both of those things are tends to working in opposite direction. To gain high efficiency, a diesel engine must be operated in relatively high load. But, on the other hands, the emission would be increased as well. Thus, it becomes one of the tasks of engineer to determine in which range of load the engine would have a high efficiency and still complies with emission regulation.

This chapter will focus to give a recommendation, based on the data that have been obtained from the engine test, about the most optimal engine operating load. Graph 17 shows the percentage of energy that converts into usefull work (brake power output). It can be seen from the graph that highest efficiency occured at 80% of engine load which has the value of 40,185% of the total energy input.

Graph 19 shows the total percentage of chemical losses at certan engine load. It can be seen from the graph that the lowest chemical losses was occured at 90% engine load. But, if we see the at the thermal efficiency, it has a lower efficiency than 80% engine load. Thus, the second alternatives is to look at the chemical losses of 80% engine load.

At 80% engine load, it has a relatively a low chemical losses. But, if we look at the NOx gases emission, it has a high production of NOx gases. Now the question is to look at the regulation whether it complies with the regulation or not. The regulation was taken from MARPOL anex VI regulation 13 about NOx requirement controls emissions from ships.

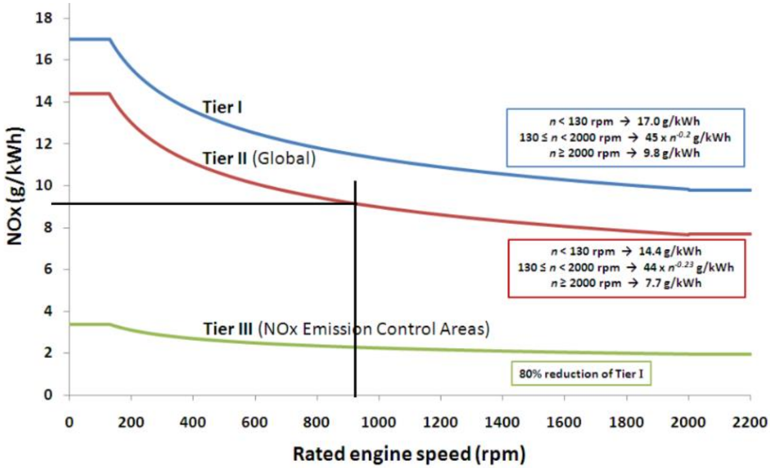
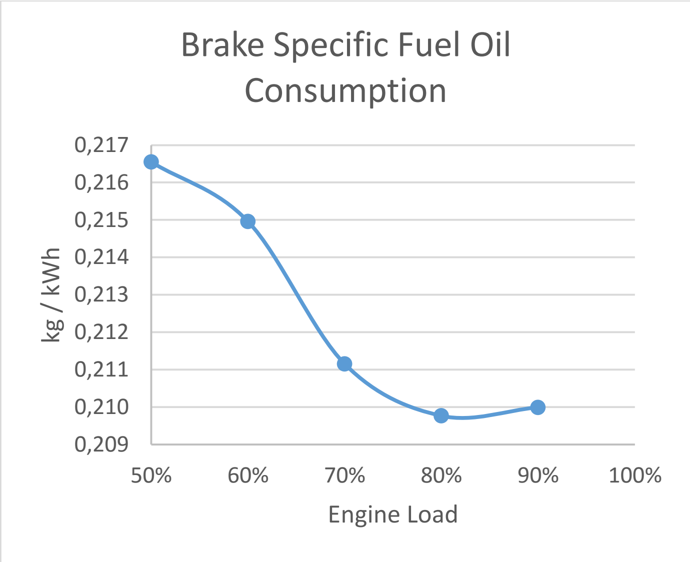


figure V.1 MARPOL NOx regulation (MARPOL Anex VI, regulation 13)

From graph 21 it can be seen that to complies NOx regulation at Tier II, the maximum NOx emission is 9,5 gr/kWH. If the value were multiplied by the engine power it becomes **8862 gr NOx / hour**. And at 80% engine load, the total NOx emission shows the value of 910,2 ppm (mg NOx / kg air). And if the value were multiplied by mass flow rates of the air it becomes **4787 gr NOx / hour**. It means that, at the 80% engine load, the NOX emissions still complies with Tier II, Anex VI, MARPOL regulation.

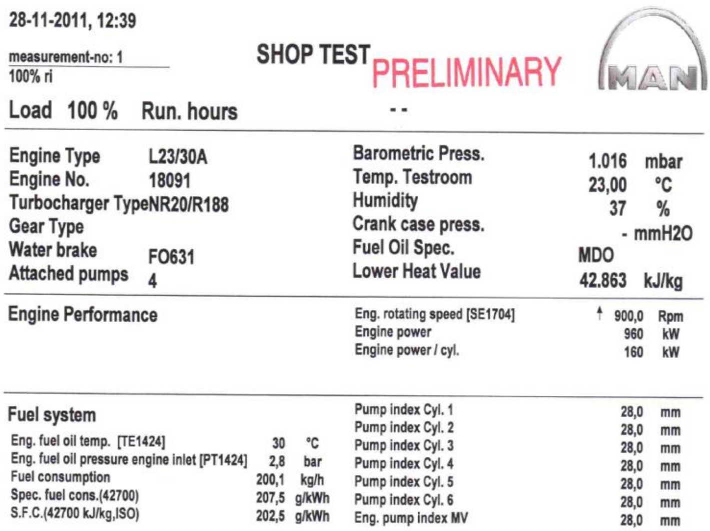


Graph V.2 Brake Specific Fuel Oil Consumption

The other important parameters to determine the engine operating area is fuel oil consumption. Graph 20 shows the brake specific oil consumption at 50% until 90% engine load. It can be seen from the graph that the lowest brake specific oil consumption occurred at 80% engine load. It indicates the value of 0,2097 kg/kwH. It might be explained because at high engine load occurred a high combustion temperature which leads to more complete combustion. Thus, it required a less amount of fuel to produce the rated power.

Thus, it can be concluded the most optimal engine operating point is 80% engine load. It has the highest brake thermal efficiency. The chemical losses has relatively a low value, and it complies with MARPOL NOx regulation. And the most important, it has a lowest brake specific fuel oil consumption.

The brake specific fuel oil consumption indicates a slightly a higher value than the value which stated on the engine shop test. The data from shop test indicates the specific fuel oil consumption with the value of 0,2075 kg/kWH (with Fuel LHV 42700 kJ/kg). So, it has a differences 0,0022 kg/kWH with the value from engine test.



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measurement-no: 1
100% ri

SHOP TEST PRELIMINARY

Load 100 % Run. hours - -

Engine Type	L23/30A	Barometric Press.	1.016 mbar
Engine No.	18091	Temp. Testroom	23,00 °C
Turbocharger Type	NR20/R188	Humidity	37 %
Gear Type		Crank case press.	- mmH2O
Water brake	FO631	Fuel Oil Spec.	MDO
Attached pumps	4	Lower Heat Value	42.863 kJ/kg

Engine Performance

Eng. rotating speed [SE1704]	900,0	Rpm
Engine power	960	kW
Engine power / cyl.	160	kW

Fuel system

Eng. fuel oil temp. [TE1424]	30	°C	Pump index Cyl. 1	28,0	mm
Eng. fuel oil pressure engine inlet [PT1424]	2,8	bar	Pump index Cyl. 2	28,0	mm
Fuel consumption	200,1	kg/h	Pump index Cyl. 3	28,0	mm
Spec. fuel cons.(42700)	207,5	g/kWh	Pump index Cyl. 4	28,0	mm
S.F.C.(42700 kJ/kg,ISO)	202,5	g/kWh	Pump index Cyl. 5	28,0	mm
			Pump index Cyl. 6	28,0	mm
			Eng. pump index MV	28,0	mm

figure V.2 Data from engine shop test

The reason beyond the differences of specific fuel oil consumption might be explained due to the leakages inside the fuel oil system. The most potential problem could occur from the internal leakages at fuel injection nozzle. The nozzle may be leaking due to a deposit build up on the nozzle or a worn out internal valve seat. In this case, fuel leaks directly into the engine’s cylinders. It would increase the fuel consumption and can result in an over-rich combustion during operation.

The other factor that could increase the specific fuel oil consumption is an external leakage. External leakage could happen along the piping circuit of fuel oil system. In general, external leakage is caused by improper assembly, loose connections, faulty gaskets, damage sealing surfaces, broken springs, engine vibration, and cracked bodies or housing. It could be repaired by tightening loose connections or replacing the damage

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AUTHOR'S BIOGRAPHY

The author was born in Jakarta, 9 November 1993, the author was the only child in his family and has taken formal education in SDK Mater Dei, SMP Pangudi Luhur Jakarta and SMA Pangudi Luhur Jakarta. The Author was graduated from SMA Pangudi Luhur in 2012, then the authors proceed to S1 or Bachelor Degree and accepted in Department of Marine Engineering Double Degree Program at Faculty of Marine Technology – Institut Teknologi Sepuluh Nopember with Hochschule Wismar Germany. The author was registered with student number NRP. 4212 101 031 in Department of Marine Engineering, the author take the Marine Power Plant (MPP) for his studies. The author was the second batch of Double Degree in Marine Engineering in Faculty of Marine Technology - Institut Teknologi Sepuluh Nopember.